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Commentary by Joseph H. Keenan

RICHARD MOLLIER

Geheimer Hofrat, Ph.D., Dr. Ing. e.h., Grashof Medalist of the Verein Deutscher Ingenieure, Professor at the Technical University of Dresden

DR. RICHARD MOLLIER, a man of quiet charm, a kindly teacher and a skillful scientist, reached the age of seventy on November 30, 1933. His name is known to engineers for his work on the properties of refrigeration fluids, for his steam tables and particularly for his invention of an engineering tool of great utility, the Mollier diagram. Within a few years of its introduction into thermodynamics Mollier's enthalpy-entropy diagram became, like the slide rule, the engineer's constant companion. The fields of power engineering and refrigeration are indebted to the author of the device which serves them day by day, and it is fitting to acknowledge that indebtedness on this anniversary.

To one who has attained only half of seventy years it seems that the literature of steam engineering can be divided quite sharply into an ancient literature and a modern literature. The two periods overlap but little and the division occurs somewhere between 1900 and 1905. The literature of the earlier period contains long, involved and now unfamiliar exposition of the properties and the "deportment" of steam. Problems in the steady flow of a fluid were the subject of circuitous discussion and complicated analysis. The more recent literature is relatively concise, coherent and lucid in its approach to engineering problems.

It is not mere coincidence that Mollier's paper on "New Diagrams for Thermodynamics" appeared during that period just preceding the modern era in steam engineering literature. Mollier's diagram was, in fact, something more than a diagram; it was a device which served to emphasize the need for a separate technique in the analysis of the all-important steady-flow problem. The steady stream of fluid was flowing through the power plants, the refrigeration systems and the steam turbines of the world of thermodynamics, but in 1900 the literature was still in terms of behavior behind a piston. The advent of the Mollier diagram made it all seem so much easier; the heat added in a boiler plant, the work done by an engine, the efficiency of a turbine, the reheat factor—all lost their esoteric qualities when exposed upon the Mollier diagram, and thermodynamics became a more useful science.

A science becomes generally useful only when it is thoroughly comprehended by a relatively large number of people. If it is understood only by a small group of specialists then the current exposition of it is poorly framed and inadequate. To expound a science well it is necessary to discern the prevailing misconceptions that hinder understanding, to eliminate from its vocabulary all unnecessary terms and to define precisely the remainder. But such refinements come late in the history of a science; they mark its transition from adolescence to maturity. A mature science recognizes its own boundaries and does not wander irrelevantly into foreign fields; it steps with dignity and grace toward its conclusions without stumbling over its own definitions; it distinguishes clearly between the questions it dares to face and those it chooses to ignore; and it has at its command useful tools which are few in number but ubiquitous in practice.

The Mollier diagram is one of these ubiquitous tools and its first appearance seems to have coincided with the coming of age of thermodynamics. It is a product of the kind of precise scientific thought which brings a science to maturity. Doctor Mollier's long career has been marked by notable contributions to scientific literature, and it has been fertile, as only the career of an inspiring teacher can be fertile, in the researches and studies of his pupils. One finds in the Mollier diagram an emblem of that career and of its devotion to the development of thermodynamics toward maturity and usefulness.

Joseph H. Keenan
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EDITORIAL

The Navy Feedwater Formula

THE Navy with its large number of ships has many more boilers than any company on land or afloat. That the ships may be ready at all times for any call it is imperative that their boilers be maintained in excellent condition. This is paramount, but the cost of repairs and renewals is, collectively, a large item.

The situation is quite different from that obtaining in land practice. Boiler water treatment and control has become a highly specialized problem. On land, feedwater supply and operating conditions vary among plants, but for the individual plant conditions are more or less definite, and the proper treatment can be prescribed by a feedwater specialist. In the Navy, however, personnel is constantly being shifted, few can be expected to possess expert knowledge and specialists are not always within reach. Consequently, the treatment must be applicable to the average case without expert supervision.

For a number of years the Navy employed a definite formula, known as "Navy Compound." This was found inadequate to meet present steaming conditions which led to an extensive series of tests and investigations being carried on at the Naval Experiment Station at Annapolis. These investigations have resulted in the adoption of a new formula, described in the A.S.M.E. paper by Messrs. Solberg and Adams, in this issue.

It is to be expected that some feedwater chemists will challenge the ability of the new formula to meet all conditions and the presentation of the paper is likely to evoke most interesting discussion. If so, the proof of their contentions must await a check-up of conditions resulting from the general application of the new formula.

Municipal Power Projects

FOR some months past news of the construction field has recorded unusual activity as to proposals for municipal power projects. These pertain to numerous small towns as well as a few cities of medium size. The issue of bonds for such work was voted down at the recent election by several cities but was carried in others.

The prospect of securing federal aid has undoubtedly motivated many of these proposals, together with the lure of cheaper electricity in lieu of existing rates. Moreover, the vicissitudes through which the central-station industry has lately passed have rendered it more vulnerable to attack by municipal plant proponents.

Granted that there are many localities where the case for the municipal plant rests on sound economics, the number of current proposals would indicate that some may have been advanced without full consideration of all the factors involved. Therefore, the requirement in the Public Works Program that such funds be advanced only for self-liquiding projects should serve as a check on hastily considered proposals.

Engineers and Employment

ESTIMATES by the U. S. Department of Labor, The American Federation of Labor and other statistical groups variously place the number of men that have been put to work as between three and four million.

These figures, however, apply almost wholly to labor and give no indication as to how the recovery program is affecting engineers. In the regrettable absence of any centralized and authoritative source of engineering employment statistics, one may glean some idea of the engineer's status from what the Engineering Societies' Employment Service and the Professional Engineers' Committee on Unemployment have found in their respective activities.

The Employment Service finds that more young men are now being placed than a year ago, at which time preference was being shown for older and experienced men. There has lately been a noticeable increase in the demand for electrical engineers and it is anticipated that when the Public Works Program gets into full swing there will be an increased demand for civil engineers. The demand for mechanical engineers is small, probably because so many positions in this class are identified with the production of capital goods, and this field has not yet felt, to an appreciable extent, the effect of governmental action.

The Professional Engineers' Unemployment Committee, which deals with the New York Metropolitan area, now has enrolled over three thousand engineers of which about fifty per cent are in urgent need of assistance. The situation appears to be more acute than it was a year ago, due to the fact that the applicants' savings are becoming exhausted. Here again, the mechanical engineers are noted to be worse off than those in the other major branches of engineering, and it is difficult to place men who are over forty, regardless of experience.

While the situation, as portrayed by the experience of these two bodies, is not bright, especially as concerns the mechanical engineer, it should not be taken as conclusive because of other factors which their records do not reveal. In the first place, there are members of the engineering societies who did not register their unemployment and consequently have failed to register re-employment. Secondly, many engineers have probably secured employment through their own efforts or through the recommendations of friends. Thirdly, as new work has developed, some firms have taken back men they were forced to let out a year or two ago. Finally, one cannot overlook the sporadic, yet highly significant, announcements of new projects and increased production schedules by important firms. These are indicative of an encouraging trend in which the mechanical engineers are almost certain to profit. As a class, however, their status is not likely to be satisfactory until capital goods are again moving. Further efforts to this end are imperative.

Heat Transfer in Air Heaters and Economizers

The subject of this paper is heat transmission in air heaters and economizers, and the matter is presented from the standpoint of a manufacturer of the apparatus who must calculate and guarantee the performance of the apparatus he is offering. The treatment of the subject is applicable to a plate- and a tube-type air heater, and to steel-tube economizers. The author's conclusions are that in figuring performance simple formulas supplemented by good judgment and comparison of the calculated results with test results, are preferable to more exact but more complicated formulas.

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AIR heater and economizer installations are usually designed for counterflow; that is, the air in the air heater or the water in the economizer flow in the opposite direction to the flow of the hot gases. Counterflow gives the greatest mean temperature difference between the hot gases and the air or water, and therefore the greatest heat transfer. The path of heat transfer consists of three parts: from the hot gases to the gas-swept surface of the metal plate, heat is transferred mostly by convection; from this gas-swept surface the heat is transferred through the metal plate to the air- or water-swept surface by conduction; and from the air- or water-swept surface the heat is transferred into the body of air or water mostly by convection.

On the gas side of the metal plate there is usually a layer of soot and ash. Adhering to this layer of soot and ash is a film of continuously changing gas. The air side surface of an air heater is generally free of any deposit but has adhering to it a film of continuously changing air. In the case of an economizer the water-swept surface may be covered with a layer of solids consisting of impurities in the feedwater. Next to this coating is a film of continuously changing water.

Heat travels by convection from the stream of hot gas into the film of gas; from this film through the layer of soot and ash and through the metal plate into the film of air it travels by conduction; and from this film of air it is taken into the moving stream of air by convection. In the case of an economizer the heat travels by conduction from the film of gas through the layer of soot and ash, the metal plate and the coating of solids into the film of water from which it is taken into the stream of water by convection.

The modes of heat transfer change in the films. On the gas side of the metal plate the flow of the gas forces into the gas film hot molecules of gas and takes out an equal number of cold molecules. The hot molecules contain large quantities of heat, part of which they impart to the film, and later they are taken out as cold molecules containing only small quantities of heat. This is convection. The faster the flow of gas over the heating surface the greater the number of hot molecules that are forced into the gas film replacing an equally large number of cold molecules, and the greater is the quantity of heat conveyed into the film.

The hot molecules that are forced into the film are in rapid motion due to their heat content. They come in contact with other molecules of the film or of the layer of soot and ash, and impart their motion to them. Thus, the heat or the molecular motion is propagated by direct contact from the gas molecules to the soot and ash molecules and by these to the metal molecules which, in turn, impart the motion to the air molecules in the air film. This impartation of the molecular motion is heat conduction. The rapidly moving hot air molecules in the air film are taken or torn out of the film by the stream of air and are replaced by an equal number of slowly moving or cold molecules. This again is heat convection. The faster the flow of air the greater is the number of cold air molecules that are forced into the air film to replace hot molecules, and the greater is the heat transfer. Similar exchange of water molecules is taking place in the water film in the economizer.

The molecules of some substances do not impart their motion to the surrounding molecules as readily as do the molecules in other substances. We say that the heat conductivity of such substances is low. Thus, the conductivity of the layer of soot and ash on the gas side surface is much lower than the conductivity of the metal of the plate, and is usually considerably lower than the conductivity of the scale on the water side of the economizer surface. In the transmission of heat under consideration we have to deal with the conductivity of the gas film, of the layer of soot and ash, of the metal and of the film of air. In the economizer we have the layer of scale and the water film. Of these the conductivity of the metal is very high and the thickness is constant. The conductivity of the gas and air films is moderately low and the thickness is nearly constant. The conductivity of the soot and ash layer is very low and the thickness is variable and difficult to estimate. The conductivity of the layer of scale in the economizer is also low, although not nearly as low as that of the soot and ash and its thickness is also variable. The low conductivity of the soot and ash layer and its variable thickness introduces an uncertain factor in the calculation of the performance of an air heater or an economizer.

Acceptance tests are usually made after the equipment has been in service two or three months and a layer of soot and ash has been deposited on its gas-swept surfaces. This and other factors discussed later demand considerable judgment in the calculations of the probable performance of an air heater or an economizer under actual operating conditions. Simple equations taking into account only the most important factors, supplemented by good judgment and comparison of the calculated results with test results of similar apparatus, are to be preferred to more cumbersome formulas which under some conditions may give more exact results.

The rate of heat transfer by convection from hot gases to the metal surface and also from the hot metal surface to the cooler air is expressed fairly accurately by the equation:

$$R = A + BM \qquad (1)$$

Where R = B.t.u. transmitted per sq. ft. of heating surface per hour per degree of temperature difference.
A and B are constants.
M = weight of gas or air per sq. ft. of cross-section area of the gas or air passage per hour, called the mass flow.

Q, the total heat transferred per hour, can be obtained by multiplying R by the effective heating surface and by the mean temperature difference, or

$$Q = RS(T_m - t_m) \qquad (2)$$

The rate of heat transfer by convection increases with the temperature of gases. However, the range of temperature variation in most air heater or economizer installations is small and usually the temperature factor can be omitted in the calculations, or can be allowed for by slightly changing the constant B in equation (1).

The rate of heat transmission by conduction is given by the conductivity of the material which is defined as the number of B.t.u. transmitted per sq. ft. of area per hour per degree difference in temperature between two parallel sections one inch apart. The quantity of heat transmitted by conduction from the gas-swept surface to the air- or water-swept surface of an air heater or an economizer is expressed by the equation:

$$Q = \frac{Ks}{D}(t_g - t_a) \qquad (3)$$

Where Q = quantity of heat in B.t.u. transmitted per hour.
K = average conductivity of the materials between the two surfaces.
D = distance in inches between the two surfaces.
s = effective surface in feet.
$(t_g - t_a)$ temperature difference between the two surfaces.

The conductivity of the metal plate and its thickness is

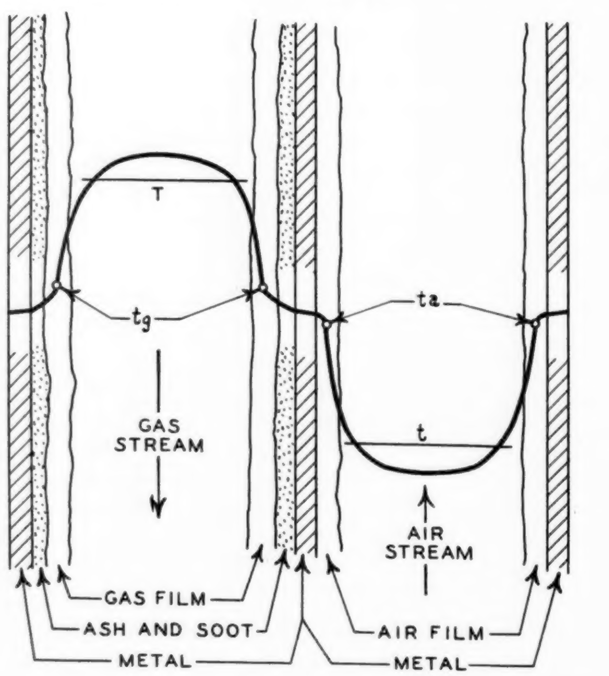

Fig. 1—Diagram of heating surfaces and relative temperatures in a plate heater

On the gas side the metal plate is shown covered with a layer of soot and ash and a film of gas. On the air side the plate has adhering to it a film of air. The curves represent the temperature of the streams of gas and air at various distances from the plate. The straight lines T and t represent the averages of these temperatures. The broken lines represent temperature gradient through the plate, layers of soot and ash and films.

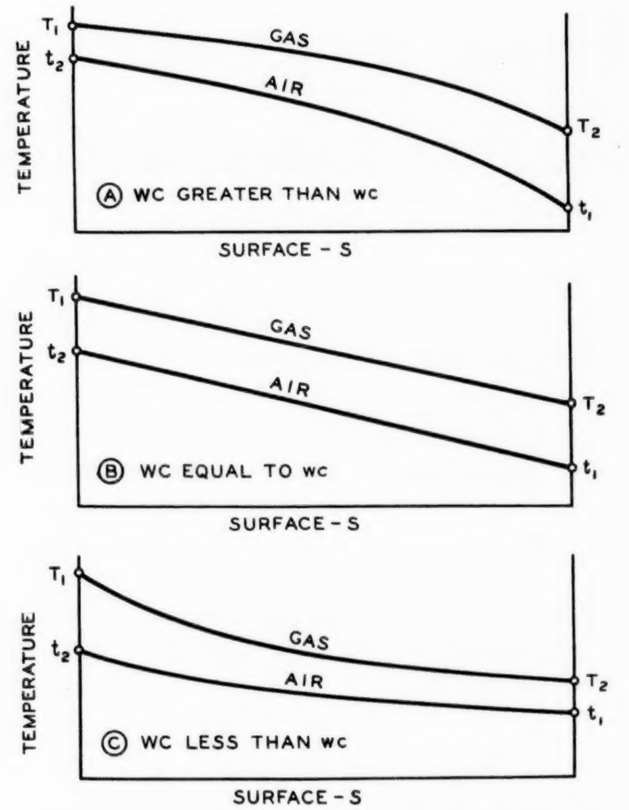

Fig. 2—Shapes and relative position of temperature curves with different ratios of $\frac{WC}{wc}$

In diagram A, *WC* is greater than *wc*; curves are concave downward and converge toward the upper end.
In diagram B, *WC* is equal to *wc*; curves become straight and parallel lines.
In diagram C, *WC* is smaller than *wc*: curves are concave upward and converge toward the lower end.

known but the conductivity of the layer of soot and ash and its thickness would have to be estimated. The conductivity and the thickness of the films of gas, air or water are also subject to uncertainty.

Assuming that the average conductivity of the materials in and on the heating plate and the total thickness can be estimated with a fair degree of accuracy, the following equation gives the quantity of heat transmitted per sq. ft. of surface per hour, from the hot gases to the air in an air heater, or, to the water in an economizer.

$$q = R_g(T - t_g) = \frac{K}{D}(t_g - t_a) = R_a(t_a - t) \qquad (4)$$

Where R_g = rate of heat transfer by convection from the hot gases to the gas-swept surface. This is computed from equation (1) in which M_g is the gas-mass flow in pounds per sq. ft. of cross-section of the gas passage per hour.
$(T - t_g)$ = temperature difference between the moving gas and the gas-swept surface.
K, D and $(t_g - t_a)$ = same as in equation (3).
R_a = rate of heat transfer by convection from the air- or water-swept surface to the moving air or water. This is computed from equation (1) in which M_a is the mass flow of air or water.
$(t_a - t)$ = temperature difference between the air- or water-swept surface and the moving air or water.

The path of the heat travel in an air heater is shown in Fig. 1 which illustrates one element with the gas stream and one element with the air stream. The metal walls are shown covered with the layer of soot and ash, and the gas and air films. The relative temperatures are shown by the curves and the

broken lines. The average temperatures of the gas and air are indicated by the straight lines T and t. At higher rates of working the broken line indicating the temperature gradient through the metal wall layer of soot and the two films is steeper and the peak of the curve representing the temperature of the stream of gas is wider. Similarly, the lowest part of the curve representing the temperature of the stream of air is wider. The part of the gas curve below the straight line T, near the wall, shows the effect of cold molecules taken out of the gas films. The part of the air curve above the straight line t, near the walls, shows the effect of hot molecules of air taken out of the air film.

M_g and M_a can be calculated from the dimensions of the air heater or the economizer, and the rate of working of the unit. Of the temperatures only T and t can be determined by measurements under actual operating condition; t_g and t_a cannot be determined without elaborate and expensive preparation which is generally impractical in any power plant.

Although equation (4) represents closely the actual process of heat transfer it is very difficult to work with and for that reason is impractical. It is, therefore, simplified and shortened by the elimination of the middle expression and by combining the first and the last expressions. The rate of heat transfer from the stream of hot gases to the stream of air in this simplified expression is given by the equation:

$$R = A + B\ M_{ga} \qquad (5)$$

Where M_{ga} = the average of the mass flow of gas and air.
A and B are constants selected to include the resistance to heat transfer of the layer of soot and ash and of the two films.

The quantity of heat transmitted per unit surface per hour is expressed by the equation:

$$q = R\ (T - t) \qquad (6)$$

Where $(T - t)$ is the temperature difference between the stream of gases and stream of air at the unit surface.

The initial temperature of the gases T_1 and the initial temperature of air t_1 entering the heater are known and the final temperatures can be computed from the equations:

$$dQ = R\ (T - t)\ ds = -\ WCdT \qquad (7)$$

$$WC\ (T_1 - T) = wc\ (t_2 - t) \qquad (8)$$

$$WC\ (T_1 - T_2) = wc\ (t_2 - t_1) \qquad (9)$$

Where WC is the weight of gases flowing through the air heater per hour times the specific heat of the gases.

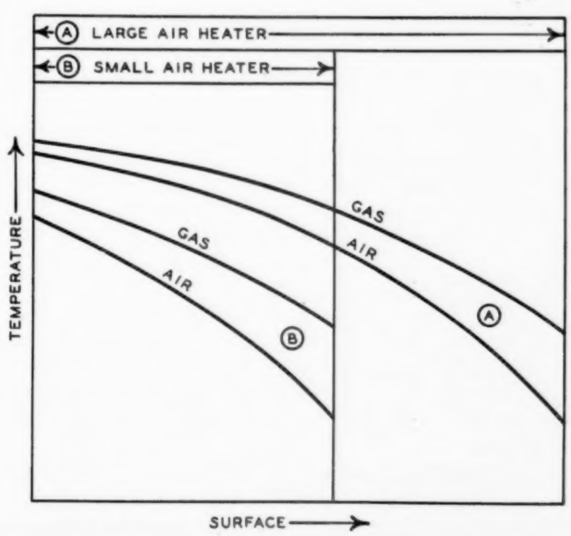

Fig. 3—Effectiveness of a large air heater with high initial gas temperature compared with the effectiveness of a small air heater with a lower initial temperature

On the large air heater the temperature difference at the hotter end is too small which greatly reduces the effectiveness of the large air heater.

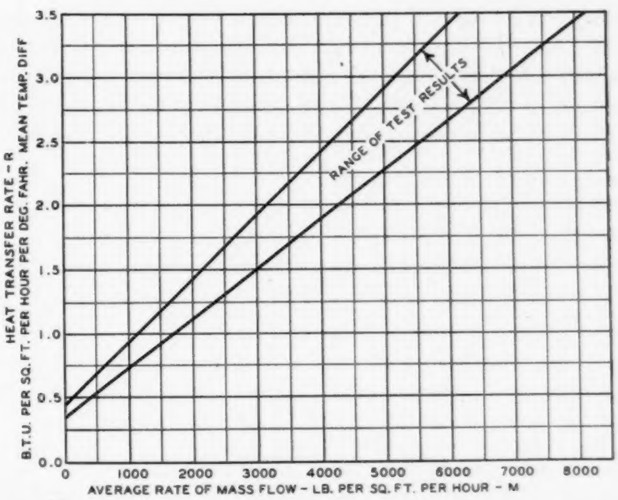

Fig. 4—Variation in the rate of heat transfer in air heaters obtained from test results

The test results fall between the two lines. The distance between the two lines represents the uncertainty of calculation.

wc = the weight of air flowing through the air heater per hour times the specific heat of the air.
T_1 and T_2 = initial and final temperature of gases.
t_2 and t_1 = final and the initial temperature of air.
T and t = temperature of the gases and air at any point in the air heater.

From (7), (8) and (9) we get

$$RS = \frac{WC}{\left(\frac{WC}{wc} - 1\right)} \log_e \frac{\left(\frac{WC}{wc} - 1\right) T_2 + t_2 - \frac{WC}{wc} T_1}{\left(\frac{WC}{wc} - 1\right) T_1 + t_2 - \frac{WC}{wc} T_1}$$

$$= \frac{WC}{\left(\frac{WC}{wc} - 1\right)} \log_e \frac{T_2 - t_1}{T_1 - t_2} \qquad (10)$$

The following equation is also true

$$RS\ (T_m - t_m) = WC\ (T_1 - T_2) \qquad (11)$$

Where $(T_m - t_m)$ is the true mean temperature difference between the gases and the air.

Dividing (11) by (10) we get

$$(T_m - t_m) = \frac{\left(\frac{WC}{wc} - 1\right)(T_1 - T_2)}{\log_e \frac{T_2 - t_1}{T_1 - t_2}} = \frac{(T_2 - t_1) - (T_1 - t_2)}{\log_e \frac{T_2 - t_1}{T_1 - t_2}} \qquad (12)$$

From the preceding equations others can be derived as they are needed. Equations (10) and (12) are applicable to cases where WC is greater than wc which is true of practically all cases of air-heater application in a power plant. When WC is equal to wc the second term of these equations becomes indeterminable. In that case the curve of the temperature of gases becomes a straight line and is parallel to the curve of temperature of air which is also a straight line; the mean temperature difference is then equal to $(T_1 - T_2)$ or $(T_2 - t_1)$ and all calculations are greatly simplified.

Fig. 2 shows the shapes and relative positions of the gas and air temperature curves when plotted on the heating surface S as abscissas. A gives the curves for cases where WC is greater than wc; B gives the curves when WC is equal to wc; and C represents the curves when WC is smaller than wc.

Cases represented by C seldom occur in power-plant air-heater installations; even cases represented by B are uncom-

mon. The weight of gases is always greater than the air used in the combustion of fuel, because to the air used in combustion is added the weight of the combustible and the moisture in the fuel. Furthermore, besides the preheated air supplied to the furnace for combustion, air which is not preheated leaks into the setting and further increases the weight of the hot gases passing through the air heater. The proportion of air leaking into the setting increases as the rate of working decreases. The specific heat of the gases on account of the CO_2 and water vapor content and the higher temperature is also greater than the specific heat of air. Generally, the value of wc varies from 0.80 WC at high ratings to 0.50 WC at low ratings. This fact of lower value of wc is well to remember when figuring the heat recovery by an air heater, especially when the temperature of gases entering the air heater is high and the temptation is to recover the heat by a large air heater.

Fig. 3 illustrates two cases; case A with high temperature of gases entering a large air heater, and case B gases at lower temperature entering a small air heater. In case A the air temperature at the higher end of the curve approaches the temperature of the gases too closely and, due to the small temperature difference, only a small amount of heat can be absorbed from the gases. A comparatively large part of the air heater adds only little to the recovery of heat.

The recirculation of a part of the heated air through the air heater increases the mass flow on the air side of the air heater and thereby raises the rate of heat transfer which nearly offsets the smaller temperature differences. It also tends to make the temperature curves two parallel straight lines. It has the advantage of preventing the sweating of the gas-swept surfaces at the point where the air enters the air heater. Sweating of the surfaces causes the dust to adhere to such surfaces and thereby reduces the heat transfer, and generally starts corrosion of the metal. Recirculation of part of the heated air is very desirable with fuel having high hydrogen content, such as oil and natural gas, and fuels high in moisture such as lignites. If recirculation is planned somewhat larger forced draft fan must be provided. At higher ratings the amount of air recirculated need be small. At lower ratings larger proportion of air recirculation can be used with benefit because the mass flow is low and there is also greater tendency for the surfaces to sweat.

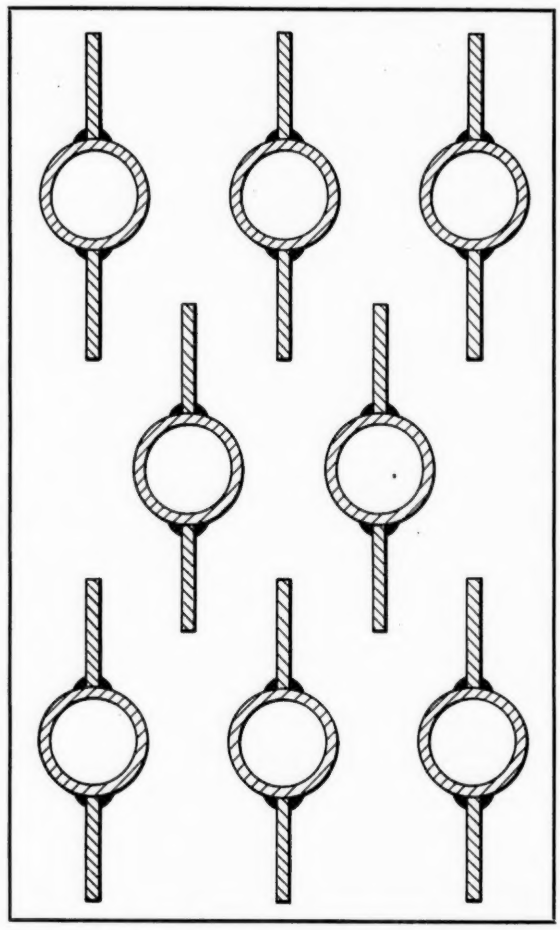

Fig. 6—Section through a part of extended surface economizer

The extended surface consists of fins welded to the tubes.

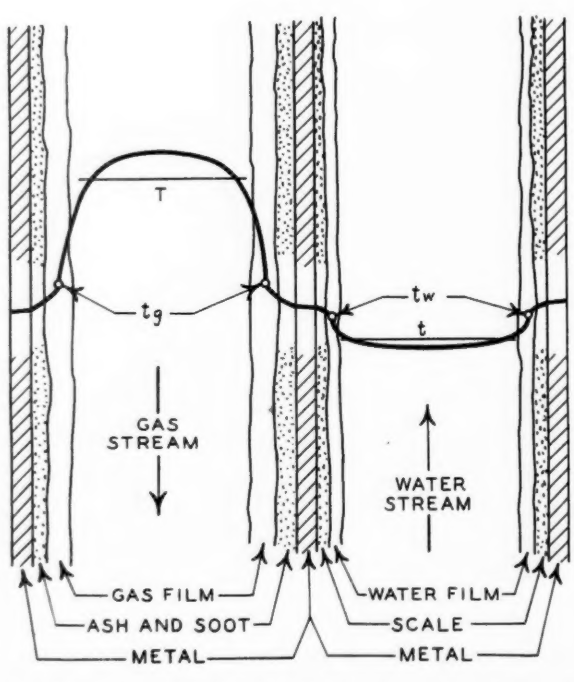

Fig. 5—Diagram shows the heating surface of an economizer covered with layer of soot and layer of scale and films of gas and water

The curves and the broken lines show the temperatures of gas, water and the heating plate with its layers of soot and scale and films. The straight lines T and t represent the average temperature of the gas and water respectively. Temperature of water is close to the temperature of the metal.

Another way of increasing the air-mass flow is to reduce the width of the air passages. Thus, in a plate heater with the gas passages 1 in. wide the air passages are made $^3/_4$ in. wide. In a tubular air heater where the air flows around the tubes, the tubes are spaced closer together.

The effective heating surface of an air heater is usually less than the total surface calculated from the air heater dimensions. There are parts of the heating surface that are not swept by the air or by the gases due to the design and arrangement of the ducts, and also to the turns in the air streams. This is especially true at low ratings when the volume of gases and air is small and does not completely fill the passages. The flow of air and gases is somewhat like the flow of water in river beds. At low flow the water does not fill the river bed completely but fills only part of the river bed and flows nearly at the same velocity as during high water time. Therefore, it is well when figuring the performance of an air heater to allow for some ineffective surface. Taking the effective surface as 90 per cent of the calculated surface brings the calculated results closer to the actual performance. With some duct and air-heater arrangements even a lower figure is advisable and may avoid later disappointment.

The cross-sectional area of the air passages in a plate-type air-heater is fairly easy to calculate. It is not so easy to calculate such areas in a tubular air heater where the cross-section of the air stream passages varies widely in short distances. Whatever cross-section is chosen it is only a nominal value and the rates of heat transfer must be adjusted to it from test results.

The above considerations indicate that in figuring the performance of an air heater simple formulas supplemented by good judgment and comparison of the calculated results with test results are preferable to more exact but also more complicated formulas. This statement does not mean that the more exact physical laws and their mathematical expression should not be studied. On the contrary, a study of the heat transfer laws in details gives the calculator a better understanding of the problem he is working on, gives him a better judgment, and enables him to make better allowances for the various uncertain factors.

For air heater calculations the average values A and B in equation (5) are 0.4 and 0.00045, respectively. In Fig. 4 the two lines give the range of the variation of these values as obtained by test results.

In the calculations of the performance of an economizer the same general scheme is used as for the performance of an air heater. In the equations small t represents the temperature of water in the economizer tubes. Since the mass-flow of water through the tubes is very high and the specific heat of water is about four times as great as that of the gases, the water absorbs readily all the heat that the gases can impart to the metal of the tubes. In other words, it is the part of the heat travel path from the stream of the hot gases to the metal of the tubes which determines the overall heat transfer rate.

Fig. 5 shows diagrammatically the relative temperature gradient from the hot gases to the water. The temperature of the stream of water t is much nearer to the temperature of the metal than is the temperature of the air in an air heater.

In equation (5) applied to economizer M is the mass flow of the hot gases. The average of the constants A and B are 2.2 and 0.00095 respectively.

The total weight of water w times its specific heat c is always greater than WC of the hot gases and, therefore, the curves representing the temperatures of gases and water are of the shape shown in case C of Fig. 2.

Inasmuch as the heat transfer on the gas side of the heating surface determines the overall rate, it is the gas-swept surface which is used in the calculation of the economizer performance. This surface is always greater than the water-swept surface, especially in the extended surface design. Such design greatly increases the gas-swept surface without increasing the water-swept surface. It puts the surface where it is most effective. The extended gas-swept surface is almost as effective as the direct surface of a plain tube economizer; it is less expensive and usually gives a somewhat lower draft loss. Fig. 6 shows a section of a part of an economizer having extended surface consisting of fins welded to the tubes. In such economizers the gases flow either up or down, and the fins give the tubes rigidity against bowing. The values of constants A and B in equation (5) as given apply to this type of economizer.

In modern power-plant practice the initial temperature of the water entering the economizer is high, 300 to 400 deg. fahr., therefore the available temperature rise is comparatively small. With such high temperature of water sweating of the gas-swept surface is eliminated and the fouling is greatly reduced. The small available water temperature rise requires small economizer which may be supplemented by an air heater.

Solubility of Sodium Sulphate and Embrittlement*

In March 1933 active work on a program of research dealing with the solubility relations of sodium sulphate in concentrated boiler-water salines was commenced under the auspices of the Joint Research Committee on Boiler Feedwater Studies. The results obtained during the first six months on the solubility of sodium sulphate in water and in solutions containing, respectively, sodium hydroxide, sodium chloride and mixtures of sodium hydroxide and chloride are presented in this paper, together with a discussion of their practical significance.

An improved type of bomb was developed from a design made available through the courtesy of W. F. Waldeck and Professor A. E. Hill, of New York University. This bomb contains its own sampler within the bomb body, obviating variations in temperature during sampling. It also is provided with a discharge device which enables the operator to procure a solid sample separated from the solution at the temperature of the experiment instead of after cooling to a lower temperature.

The air thermostat constructed for the solubility tests maintains a temperature at all points in its interior constant within 0.5 cent. (0.9 fahr.). To obviate danger, sampling of bomb solutions is conducted without opening the thermostat.

The solubility of sodium sulphate in water was checked at temperatures ranging from 150 to 350 cent. (302 to 662 fahr.). From 150 to 250 cent. the solubility does not vary greatly from an average of 43 grams of Na_2SO_4 per 100 grams of H_2O. Above 250 cent., however, the solubility decreases sharply and apparently approaches zero at 374 cent. (706 fahr.).

At any temperature from 150 to 250 cent. (302 to 482 fahr.) the solubility of sodium sulphate decreases with increasing concentration of sodium hydroxide. For any particular NaOH concentration, however, the solubility of sodium sulphate increases with increase in temperature over the range specified. Thus, while the solubilities at 150, 200 and 250 cent. are nearly the same in pure water, in a solution containing 10 grams of NaOH per 100 grams of H_2O they are, respectively, about 25, 30 and 35 grams of Na_2SO_4 per 100 grams of H_2O.

The effect of sodium chloride upon the solubility of sodium sulphate has never been considered in the discussion of embrittlement ratios. In the present work, however, it has been found that sodium chloride lowers the solubility at 150 cent. (302 fahr.) just as much as does sodium hydroxide and surpasses this substance in its depressing effect at 200 and 250 cent. (392 and 482 fahr.). For example, in a solution containing 10 grams of NaCl per 100 grams of H_2O the solubilities at 150, 200 and 250 cent. are, respectively, about 26, 29 and 28 grams of Na_2SO_4 per 100 grams of H_2O.

At a temperature of 300 cent. (572 fahr.) a form of sodium sulphate is encountered which differs essentially in its behavior from the forms present at the lower temperatures. At 300 cent. the solubility of sodium sulphate increases with increase in NaOH concentration although it decreases with increase in NaCl concentration. In solutions containing 10 grams of added substance per 100 grams of H_2O, the solubilities at this temperature in NaOH and NaCl, respectively, are about 32 and 18 grams of Na_2SO_4 per 100 grams of H_2O.

In solutions containing both NaCl and NaOH in varying concentrations the behavior of sodium sulphate at 150, 200 and 250 cent. is very nearly that which would be estimated from the determinations in each of these substances separately. While all mixtures with the same total concentration of sodium hydroxide and sodium chloride are practically equivalent in depressing the solubility of sodium sulphate at 150 cent. (302 fahr.), an increase in the ratio of NaCl to NaOH decreases the solubility at 200 and 250 cent. (392 and 482 fahr.).

No attempt can be made at present to evaluate control ratios. Curves are presented, however, to show the effect of temperature, NaOH concentration and NaCl concentration on the value of the Na_2SO_4/NaOH ratio corresponding to equilibrium with solid sodium sulphate. The value of this ratio at any temperature decreases rapidly with increase in NaOH concentration and with increase in the ratio of NaCl to NaOH.

* Abstract of paper by W. C. Schroeder, Research Chemical Engineer, and Everett P. Partridge, Supervising Engineer, Non-Metallic Minerals Experiment Station, U. S. Bureau of Mines, New Brunswick, N. J. Presented at the Annual Meeting of The American Society of Mechanical Engineers.

The Thermal Performance of the Detroit Turbine Using Steam at 1000 Deg. Fahr.

This paper[1] gives the results of a large number of tests made to ascertain the thermal performance of a 10,000-kw. turbine-generator designed for 1000-deg. steam. The thermal efficiency was found to be 31.8 per cent and the engine efficiency 76 per cent for the complete unit. A companion A.S.M.E. paper by P. W. Thompson and R. M. Van Duzer, Jr., giving experiences with the generation and utilization of the steam at this temperature, with special reference to the materials employed, appeared in full in the November issue of COMBUSTION.

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THE advantages of using high-temperature steam at the throttle of a turbine, as compared with low-temperature steam of the same pressure, used without reheating, are: (*a*) more energy is available for transformation into work for any specified exhaust pressure; (*b*) a larger portion of this available energy can be utilized; and (*c*) less erosion of the turbine blades in the low-pressure stages is produced. These advantages are of sufficient importance in the operation of large central stations to justify the expenditure of large sums of money to ascertain new facts concerning the use of high-temperature steam. Even though steam temperatures above 1000 deg. fahr. have been previously used in certain static apparatus, the problems encountered in designing turbine parts to sustain high temperatures are much more difficult, because small clearances between the high-speed rotor and the stationary elements must be maintained in spite of the tendency of the materials to grow and become seriously distorted when heated nearly 1000 degrees above that prevailing when the machine is cold.

The main results obtained from the analysis of the tests of the Detroit unit may be briefly expressed as follows:

(*a*) The energy consumption of the complete unit (turbine-generator and three heaters) was 10,730 B.t.u. per kw. hr., for a load of 10,000 kw., a throttle pressure of 390 lb. per sq. in. abs., a throttle temperature of 1000 deg. and an exhaust pressure of 1 in. Hg abs.; this means a thermal efficiency of 31.8 per cent, and an engine efficiency of 76 per cent for the complete unit.

(*b*) A load of 10,000 kw. with 1000-deg. steam was not large enough to give the highest thermal efficiency of this unit.

(*c*) Increasing the steam temperature from 700 to 1000 deg. reduced the energy consumption of this unit 920 B.t.u. per kw. hr., or 7.9 per cent.

(*d*) The radiation and convection losses from the turbine and heaters with 1000-deg. steam were relatively small, namely, 0.6 per cent of the available energy for a load of 10,000 kw.

(*e*) The loss due to the leakage of sealing steam was relatively large, namely, 4.4 per cent of the available energy for full-load conditions; this is probably not an inherent characteristic of large turbines designed for 1000-deg. steam.

(*f*) The results of these tests become attractive when viewed from the future possibilities of 1000-deg. steam used in large units with a far smaller percentage of loss due to the sealing steam, since the complete elimination of this loss (not an inconceivable attainment in a unit of 50,000 kw. or larger) would mean an energy consumption of about 9900 B.t.u. per kw. hr. of net generator output for the same steam pressures as used in these tests. For a pressure of 1200 lb. per sq. in., and no reheating, one may reasonably expect that a very large unit (say 75,000 kw. or more) would require an energy consumption of about 8600 B.t.u. per kw. hr.

(*g*) The foregoing conclusions refer only to thermal efficiencies. The authors are not overlooking the difficulties of design and of material that have arisen in the building of the present small machine; neither are they failing to recognize that opinions expressed on behalf of the turbine manufacturers that such a large machine as that described in (*f*) can be built, are implicit rather than explicit. The authors have, in the text of the paper, made some suggestions as to modifications of design, and inasmuch as they are neither designers nor builders of turbines, they do not wish to expand these. But they take occasion, nevertheless, to say that in their opinion the difficulties of building such a large machine for steam having a temperature of 1000 deg. are not insuperable.

(*h*) The over-all coefficients of heat transfer in the feedwater heaters that received highly superheated steam in these tests appear to be very high relative to those obtained when saturated steam is used; but these are merely nominal values based upon nominal mean temperature differences rather than upon the real ones. The temperatures of the feedwater leaving the heaters, as indicated by the nominal temperature differences which were based upon the saturated temperatures of the entering steam, approached closely the values applying to the ideal unit herein used. On the other hand, the true terminal temperature differences of the heaters were much larger than the nominal as they were based upon the actual temperature of the highly superheated steam entering the heaters.

The most important data pertaining to the tests with 1000-deg. steam and the results obtained therefrom are summarized

[1] Prepared, under the auspices of the Power Division, for presentation at the Annual Meeting of the American Society of Mechanical Engineers, New York, December 4 to 8, 1933.

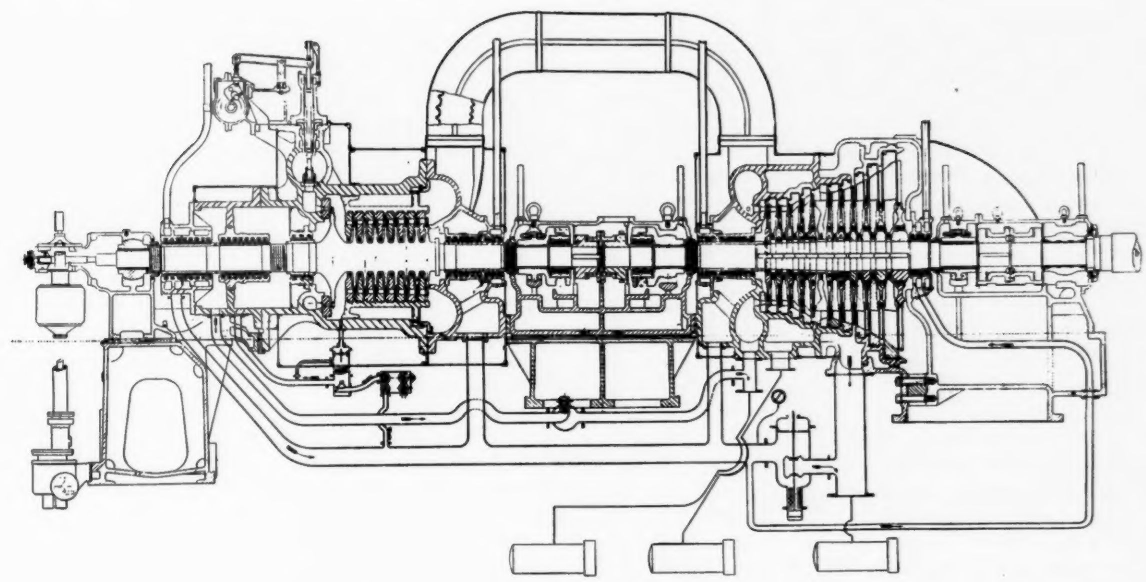

Fig. 1—Sectional view of the 1000-deg. turbine

in Table I. Attention is called to the two items for the steam rate as given in this table. The sum of these two steam rates is purposely omitted, as such a result would be of little significance, since the energy supplied with each pound of throttle steam was much greater than that supplied per pound of sealing steam.

Description of the Unit

The unit tested consists of a horizontal tandem-compound turbine, a 12,500-kv.a. generator, and three regenerative feedwater heaters. The turbine was designed for a steam pressure of 380 lb. per sq. in. abs. at the throttle, an exhaust pressure of 1 in. Hg abs., and a steam temperature of 1000 deg. at the throttle. It operates at 3600 r.p.m., and is of the impulse type with nine stages in the high-pressure cylinder and eleven in the low. The generator operates at 4800 volts and 60 cycles. A cross-section of the turbine is shown in Fig. 1. The pitch diameters of the blading in the first, second, ninth, tenth and twentieth stage wheels are approximately 32½, 23, 23½, 32 and 50 in., respectively. The oil pump, exciter and generator ventilating fans are driven by the turbine; hence the output of the generator as given under the heading "Load" in the test data represents the net output of the unit, except for the negligibly small (0.1 per cent) amount of energy expended to operate the hotwell pump and the heater-drains pump. The feedwater heaters are of the horizontal four-pass type with heating surfaces as given in Table VI. The drains from the heaters are handled as shown by the diagrammatic sketch in Fig. 2. The exhaust steam from the turbine is delivered to an 8000-sq. ft. two-pass condenser, which is slightly undersized for this service, as it was originally intended for another installation.

The turbine speed is controlled by a flyball-type governor driven directly from the turbine shaft, and the governing is accomplished by means of five control valves that are operated by a camshaft moved by an oil servo-motor controlled by the governor.

The high-pressure shaft packing constitutes an important part of the turbine because leakage of the high-temperature steam might cause serious heating of the shaft and bearing in addition to the loss of energy due to the leakage. The axial length of the high-pressure packing is greater than the assembly of the first nine wheels, as may be seen in Fig. 1. This extended form of the packing permits the shaft to be kept reasonably cool by the sealing steam leaking through the packing, and thus the main bearing is not endangered by the high steam temperature at the throttle. The steam leaving the first-stage wheel has a temperature of about 930 deg. and is kept from passing through the packing glands by using saturated steam of slightly higher pressure as sealing steam. When the sealing-steam regulator is set to give a pressure at entrance to the high-pressure packing of 0.5 lb. per sq. in. above that in the first-stage shell, the leakage of saturated steam past the inner packing and into this shell amounts to about 10 per cent of the total sealing steam, or about 0.6 per cent of the total

TABLE I

SUMMARY OF TEST AT 1000 F

1. NET LOAD, at generator terminals, kw	10,068	8,034	5,986	4,042
2. TEMPERATURE, F				
(a) Of steam entering throttle	1,003	1,006	1,004	1,005
(b) Of sealing steam entering regulator	445	448	450	452
(c) Of feedwater leaving 9th-stage heater	337	322	304	282
3. ABSOLUTE PRESSURE, lb per sq in.				
(a) Of steam entering throttle	392	390	393	387
(b) Of sealing steam entering regulator	391	405	415	420
(c) Of steam at 9th-stage bleeder nozzle	128	103	78	55
(d) Of steam at 14th-stage bleeder nozzle	40.5	32.5	24.3	17.4
(e) Of steam at 17th-stage bleeder nozzle	10.5	8.4	6.3	4.5
(f) Of steam entering condenser	0.50	0.49	0.46	0.52
4. STEAM RATE, lb per kwhr				
(a) To throttle	8.388	8.403	8.519	8.965
(b) To sealing-steam regulator	0.541	0.530	0.523	0.559
5. GENERATOR EFFICIENCY, per cent (Including bearings, ventilation, and excitation; furnished by B.T.H. Co.)	95.58	95.07	94.11	91.90
6. ENERGY CONSUMPTION RATE, Btu per kwhr				
(a) Of complete unit (turbine-generator and heaters)	10,731	10,890	11,186	11,995
(b) Of turbine and heaters	10,257	10,353	10,527	11,023
7. THERMAL EFFICIENCY, per cent				
(a) Of complete unit	31.81	31.34	30.51	28.45
(b) Of turbine and heaters	33.28	32.97	32.42	30.96
(c) Of corresponding ideal unit	41.86	42.00	42.28	41.59
8. ENGINE EFFICIENCY, per cent				
(a) Of complete unit	75.99	74.62	72.16	68.41
(b) Of turbine and heaters	79.50	78.49	76.68	74.44

Editor's Note: The abbreviations appearing in these tables, which were reproduced directly from the author's manuscript, agree with the Tentative Standard Abbreviations of the American Standards Association, whereas those appearing in the text have been edited to conform to the style manual of this magazine.

flow to the turbine, for a load of 10,000 kw. Part of the sealing steam from the intermediate high-pressure packing passes on to the outer packing and the remainder goes to the ninth-stage heater. Part of the leakage through the outer packing seals the shaft packing on the exhaust end of the turbine, and the remainder passes through a pressure regulator on its way to the seventeenth-stage heater. The connecting lines between the turbine, heaters, pumps, and seals are shown in Fig. 2. The atmospheric vents, marked *A* in the diagram, discharge very small amounts of vapor, and thus serve to give constant evidence to the operator that all seals are being properly maintained. No water seals are used on any of the shaft packings. The packing is of the saw-tooth type, as shown in Fig. 3, and is arranged in the form of individual rings, each of which is composed of four sectors. The sectors are held in place by leaf springs, which are intended to provide sufficient flexibility to prevent damage to the packing from small shaft deflections that are likely to occur during starting.

The high-pressure cylinder has an inner and an outer casing, as shown in Fig. 1. The inner casing holds the eight interstage diaphragms, and is made in two halves with axial flanged joints. The outer casing supports the inner one, fitting it snugly at the end nearer the throttle, and has only circumferential joints at its two ends. This construction eliminates the junctions of axial and circumferential joints used in the conventional design turbine cylinders with split casings, which might be difficult to maintain in a tight condition with steam at such a high temperature.

The materials used in the construction of the turbine are fully discussed in the contemporary paper on "High-Temperature Steam Experience at Detroit," by P. W. Thompson and R. M. VanDuzer, Jr.

The turbine and generator were made by The British Thomson-Houston Company, Ltd., of Rugby, England, and the heaters were made by The Griscom-Russell Company, of New York City. The unit was installed in Delray Power House No. 3, of The Detroit Edison Company in 1930. Before the tests were made the unit had been operated under load for more than 4700 hours, of which 1500 had been with steam at 1000 deg. This operation involved a large number of separate runs, which caused noticeable wear of the high-pressure packing glands during the starting periods. Although the glands were refitted after this operation, some wear was undoubtedly caused by starting the unit eight times before the final tests were made.

Test Procedure

Forty runs were made on this unit under various conditions, such as load, temperature of steam at the throttle, temperature of sealing steam and exhaust pressure. Only twenty-four runs are tabulated in this paper as they are the ones for which the operating conditions were the nearest to those for which the unit was designed.

The loads on the unit were nominally 10,000, 8000, 6000 and 4000 kw. The pressure of the steam at the turbine throttle was maintained at approximately 390 lb. per sq. in. abs. by manual operation of a throttling valve at the superheater outlet. This throttle pressure was about 3 per cent above that for which the turbine was designed. The temperature of the steam at the turbine throttle was maintained at 1000 deg. during the principal runs and at 900, 800 and 700 deg. on the supplementary runs. The steam supplied to the throttle was station steam of approximately 700 deg. that had been further superheated in an oil-fired superheater.

During most of the runs saturated steam at a pressure slightly above that in the first-stage shell of the turbine was used to seal the high-pressure shaft packing. In the earlier runs this differential pressure was about 1.5 lb. per sq. in. and was gradually reduced to 0.5 as the runs progressed. Certain runs were repeated in order to study the effect of using sealing steam at 700 deg.

The exhaust pressure during some of the runs was undesirably high as they were made in the summer time with warm condensing water and no attempt was made to regulate the exhaust pressure to some constant value. While the runs in the winter time were being made, the exhaust pressure was regulated to approximately 1 in. Hg abs. for all loads by admitting air to the vacuum pump suction. During all runs the power factor was maintained at the design value of 80 per cent.

Each run was of such duration that approximately 40,000 kw. hr. were generated. This insured uniform accuracy in weighing condensate and measuring electrical output in all runs. The frequency of making instrument observations was adjusted on the runs of various lengths so that there would be approximately the same number of observations on each run. Portable telephones were used to synchronize the essential observations. The personnel consisted of eleven men.

The pressures of the steam at various locations were measured with different types of instruments. Those of the steam at the throttle, at the sealing steam inlet, and at the higher pressure bleeder nozzles were measured with Bourdon spring pressure gages. The pressure at the low-pressure bleeder nozzle was obtained by means of a mercury manometer. The exhaust pressure was measured by two barometer-type mercury columns connected to tapped openings in opposite sides of the distance piece between the turbine exhaust opening and the condenser. These two openings were in a vertical plane approximately 2 ft. beyond the last-stage wheel. Atmospheric pressure was measured by a mercury barometer.

Method of Measuring Temperatures. The temperatures of the steam entering the throttle and of the sealing steam were measured by means of a potentiometer-type thermocouple system. Iron-constantan thermocouples were peened into the outside surfaces of the pipe, which were well insulated. The wires of the thermocouples were electrically insulated from the pipe and were wrapped around the pipe before being led out through the pipe covering. The high accuracy of this method of measuring the temperature of steam flowing in a well-insulated pipe results from the very small temperature gradient from the flowing steam to the outside surface of the pipe. This temperature head is small, as there is a very small amount of heat conducted through the liberal thickness of pipe covering. The relative accuracy of this method of measuring the temperature of steam compared with that of using a thermocouple welded into the bottom of a thermometer well is indicated by the curves in Fig. 4. This comparison, which is one of several made by the Research Department of The Detroit Edison Company, is at a somewhat lower temperature than the maximum temperature of 1000 deg. encountered with this turbine, but these curves show that this method of measuring the temperature of steam in a well-insulated pipe is reliable.

All other temperature measurements except one were made with etched-stem thermometers. The exception was that of the steam entering the ninth-stage heater which was measured with a thermocouple inserted in a thermometer well because of insufficient room to insert an etched-stem thermometer.

Steam Consumption Measurements. Measurement of the steam consumption was complicated because the amount of condensate from the hotwell, the sealing steam, the bled steam, the leak-off steam and leak-off sealing water had to be made separately. The water for sealing the shaft glands of the pumps and the relief valves on the condenser and heaters was condensate from this unit.

During the tests with steam at 1000 deg., the condensate from the hotwell, instead of being pumped directly to the low-pressure heater, as shown in Fig. 2, was delivered by two pumps in series to open weighing tanks. From these tanks it flowed by gravity to an open dump tank, through a deaerator, through a surface-type cooler and to the suction of the hotwell pump shown in Fig. 2. Thus, it started through the feedwater heaters at substantially the same temperature that it had at the hotwell outlet. The weighing tanks were provided with leak-proof dump valves and with large certified calibration weights which could be applied at will by means of hydraulic jacks.

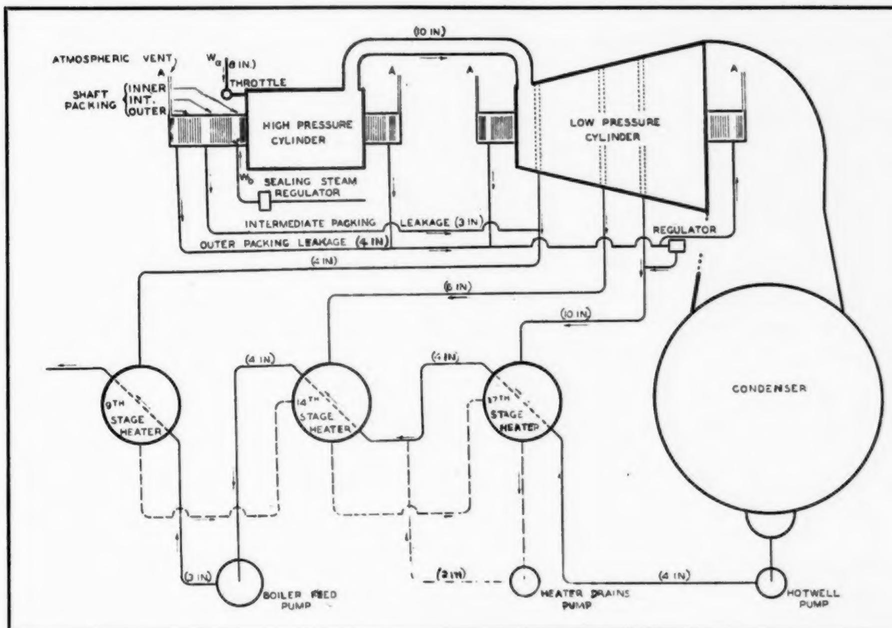

Fig. 2—Diagram of turbine and heater connections

The rates of flow of sealing steam to the high-pressure shaft packing and of the leak-off from this packing to the low-pressure shaft packing and heaters were measured by means of flow nozzles and differential manometers. Mercury under water was usually used in these manometers, but in order to increase the differential indicated by the manometers in some cases acetylene tetrabromide was substituted for the mercury. Its specific gravity is 22.1 per cent of that of mercury at 32 deg.

The steam bled from the turbine for heating the feedwater was supplemented in two of the heaters by leak-off sealing steam from the shaft packings, as shown in Fig. 3. The condensate produced in the heaters was cascaded successively from the highest pressure heater to the lowest pressure one. From the latter it was pumped into the feedwater circuit beyond the outlet of the lowest pressure heater. The quantity of steam bled from each extraction nozzle was calculated from the energy-balance of the three heaters. The known data entering into this energy-balance were, (1) the rate of flow of feedwater through the low-pressure heater, (2) the rate of sealing steam leak-off from the high-pressure shaft packing to the heaters, (3) the pressure of the steam entering each heater, and (4) the temperatures of the heater drains entering the water circuit beyond the low-pressure heater, of the feedwater entering and leaving each heater, and of the steam entering each heater.

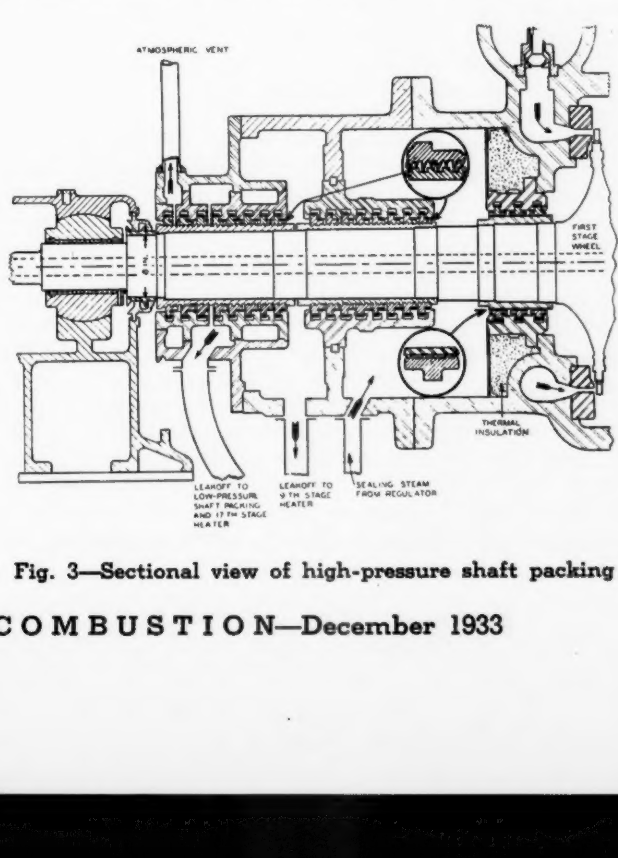

Fig. 3—Sectional view of high-pressure shaft packing

The small amount of sealing steam escaping from each of the four shaft-packing atmospheric vents was measured once during the tests by collecting the steam in a pipe leading to a small condenser. The weight of the condensate formed during a definite period determined the rate of flow, which was assumed constant for all runs.

The steam leak-off from the throttle valve and the control valves was condensed and, being unmeasured, was admitted to the suction of the pump that delivered the hotwell condensate to the weighing tanks.

All of the water used for sealing the relief valves on the heaters, as well as on the turbine exhaust, and for sealing the shaft glands on the pumps was condensate extracted from various points in the water circuit of the unit. In some cases the water had not yet reached the weighing tanks before it was withdrawn to be used for sealing purposes, while in the other cases it had been through the weighing tanks before being diverted from the main water circuit. In the former cases the leak-off was collected in tanks and was periodically siphoned into the suction of the pump that delivered the hotwell condensate to the weighing tanks. In the latter cases the leak-off was measured in open tanks and was then discarded; but its magnitude was not sufficient to affect the heater performance whether it passed through the heaters or not. Although the leak-off in the latter cases was small, it was applied as a correction to the weighed water to determine the amount of feedwater entering the low-pressure feedwater heater.

The piping system for this unit was thoroughly isolated from the rest of the plant at all connecting points, by means of double valves with open drips between them.

Tests for leakage of condensing water into the steam space of the condenser, and of cooling water into the weighed condensate as it passed through the cooler, which were continually made by the electrical conductivity method, showed that there was no inleakage of water that was detectable. A careful test of the entire circuit through which condensate from both sources of steam passed was made. It showed that no steam of condensate escaped measurement. The errors in determining the entire steam flow very probably did not exceed plus or minus 1 per cent.

Generator Output. The generator output was measured by means of instrument transformers and an integrating wattmeter which had a low multiplier, namely, 100.

Radiation and Convection Losses. During one run with

steam at 1000 deg. and a load of 10,000 kw., sufficient data were taken to calculate the heat transmission from the unit by radiation and convection. These data included, (1) the area of surfaces of the turbine, of the feedwater heaters, of the steam-extraction lines, and of the water piping between heaters, (2) the temperatures of these surfaces, (3) the ambient air temperature, and (4) the temperatures of other bodies that absorbed radiation from the unit. The only parts of the unit that were not well insulated were the bonnets of the turbine control valves and the pumps.

Calibration of Instruments. All of the instruments used in these tests were carefully selected for their appropriateness and reliability. They were calibrated with care both before and after the tests. Some of them were occasionally given additional checks between runs.

Alternate Test Procedure in Supplementary Runs. In the supplementary runs, during which the temperature of the steam at the throttle was 900, 800 or 700 deg., all measurements were made as in the runs with steam at 1000 deg., except that the hotwell condensate was not weighed, but was pumped directly to the feedwater heaters as during normal operation. A carefully calibrated integrating venturi meter on the discharge side of the high-pressure heater measured the hotwell condensate plus the heater-drains.

The probable errors in determining the entire steam flow in these supplementary runs were not more than plus or minus 1.5 per cent.

Engine Efficiency and the Ideal Unit

The thermal performances of turbines and turbine-generators may be expressed in terms of their thermal efficiencies, engine efficiencies, and rates of energy consumption per unit of mechanical or electrical energy delivered by the unit. Each of these forms of expression is useful, and the one of most value depends upon the purpose for which the performance data are desired. With a turbine that uses steam of higher temperature or higher pressure than is usual, however, the engine efficiency becomes of the greatest interest because this term expresses more satisfactorily than any other one the degree of perfection attained by such a turbine. The engine efficiency of any unit is defined as the ratio of the thermal efficiency of the actual unit to that of the corresponding ideal unit. The word "unit" in this definition may be taken to mean the turbine and its regenerative feedwater heaters, or the turbine-generator and heaters.

All real turbines have the following imperfections: throttling at admission, leakage at the high-pressure and low-pressure packings, leakage between stages, nozzle and blade losses, rotational losses, exit velocity loss, bearing friction, and heat transmission from all external parts of the machine to its surroundings. The ideal turbine has none of these losses because it is defined as a machine that can utilize all of the available energy of the steam for any specified throttle condition, exhaust pressure, and any given number of feedwater heaters extracting steam at the same pressures as those used in the actual turbine.

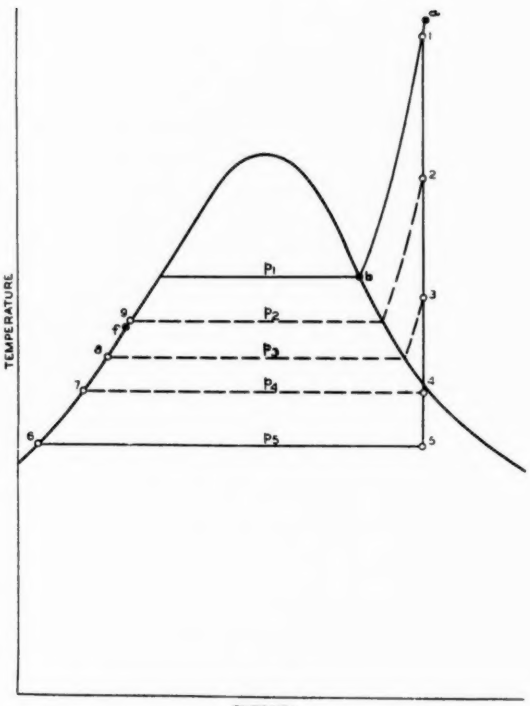

Fig. 5—States of the fluid in the ideal unit

When a turbine bleeds steam to heat feedwater the combination of the turbine and its heaters constitutes a regenerative unit, and the corresponding ideal unit may include an infinite number of heaters or the same number of heaters as are employed in the actual case. The choice of which ideal regenerative unit to select as a standard on which to base the engine efficiency of the actual unit depends chiefly upon the purpose for which the results are needed. For those cases, however, in which a specific turbine has been operated and tested with a finite number of heaters in use, there is one important advantage in comparing its performance with that of an ideal unit having the same number of heaters as the actual. With this procedure the effect of the number of heaters in use on the engine efficiency is eliminated; and the engine efficiency of the turbine and heaters will, therefore, be less than 100 per cent simply because of the imperfections in the turbine and heaters.

The actual heaters have certain losses, such as those due to throttling of the steam in its passage from the turbine to the heaters, heat transfer from the outside of the heater to the surroundings, and a terminal temperature difference. These heater losses are not large, but nevertheless they tend to make the engine efficiency of the regenerative unit somewhat less than that of the same unit operated on the Rankine cycle. However, the efficiency of the ideal regenerative unit with a finite number of heaters is generally sufficiently higher than that of the Rankine to make the thermal efficiency of the actual regenerative unit far superior to that obtained without regenerative feedwater heating. In this connection, attention is called to the bare possibility of having a higher engine efficiency with the regenerative unit than with the same turbine operating on the Rankine cycle, because it is possible, at full load or overload, to have the blades in the last stage of a turbine too small to handle efficiently the total throttle flow, and thus the last-stage leaving loss may become sufficiently large to counteract the effect of the heater losses in the regenerative unit.

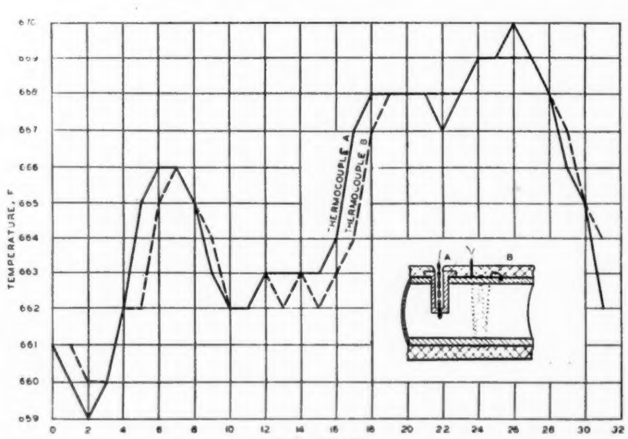

Fig. 4—Comparison of thermocouple readings resulting from two different methods of installation

TABLE II

DATA FOR CALCULATING THE EFFICIENCY OF THE IDEAL REGENERATIVE UNIT

Run No.	Date	Load, kw (L)	Throttle Steam: Pressure, lb per sq in. abs (p_a)	Throttle Steam: Temp., F (t_a)	Throttle Steam: Enthalpy, Btu per lb (h_a)	Throttle Steam: Flow, lb per hr (w_a)	Sealing Steam: Pressure, lb per sq in. abs. (p_b)	Sealing Steam: Temp., F (t_b)	Sealing Steam: Enthalpy, Btu per lb (h_b)	Sealing Steam: Flow, lb per hr (w_b)	Mixture: Enthalpy, Btu per lb (h_1)	Mixture: Entropy, Btu per F per lb (s_1)	Mixture: Flow, lb per hr (w_1)	Condition of Steam in Ideal Turbine: 9th Stage Extraction Nozzle: Pressure, lb per sq in. abs (p_2)	9th Stage Extraction Nozzle: Enthalpy, Btu per lb (h_2)	14th Stage Extraction Nozzle: Pressure, lb per sq in. abs (p_3)	14th Stage Extraction Nozzle: Enthalpy, Btu per lb (h_3)	17th Stage Extraction Nozzle: Pressure, lb per sq in. abs (p_4)	17th Stage Extraction Nozzle: Enthalpy, Btu per lb (h_4)	Exhaust: Pressure, in. Hg abs (p_5)	Exhaust: Enthalpy, Btu per lb (h_5)	Enthalpy of Saturated Liquid Corresponding to: Exhaust Pressure, Btu per lb (h_6)	17th Stage Pressure, Btu per lb (h_7)	14th Stage Pressure, Btu per lb (h_8)	9th Stage Pressure, Btu per lb (h_9)	Efficiency of Ideal Regenerative Unit, per cent (e_i)
												Regulated Exhaust Pressure of Approximately 1.0 in. Hg abs														
1	12-14-32	10068	392	1003	1529	84448	391	445	1206	5444	1509	1.756	89892	129	1355	40.5	1238	10.5	1126	1.012	945	48	163	237	317	41.86
2	12-13-32	8034	390	1006	1530	67512	405	448	1206	4256	1511	1.759	71768	103	1332	32.5	1215	8.4	1111	0.997	945	47	153	223	300	42.00
3	12-13-32	5986	393	1004	1529	50996	415	450	1205	3130	1510	1.756	54126	77.9	1298	24.3	1189	6.3	1091	0.938	939	45	140	207	280	42.28
4	12-14-32	4042	387	1005	1530	36237	420	452	1206	2259	1511	1.760	38496	55.2	1267	17.4	1166	4.5	1071	1.059	949	49	126	189	256	41.59
5	1-10-33	10063	389	700	1363	106691	395	452	1211	5780	1355	1.637	112471	142	1247	44.9	1149	11.9	1056	0.975	878	46	170	243	326	39.98
6	1-11-33	8045	388	703	1364	85088	414	456	1210	4059	1357	1.639	89147	112	1226	35.3	1133	9.5	1043	0.957	878	46	158	228	307	39.90
7	1-12-33	6036	389	701	1363	64587	425	459	1211	2856	1357	1.639	67443	85.1	1202	26.9	1112	7.2	1025	0.995	880	47	146	213	286	39.68
8	1-13-33	4038	390	703	1364	44602	428	459	1211	2116	1357	1.638	46718	59.2	1172	18.4	1085	5.0	1003	0.986	878	47	130	192	261	39.78
												Unregulated Exhaust Pressure.														
9	8-24-32	9960	389	1003	1529	86030	396	445	1205	4978	1511	1.758	91008	129	1359	40.8	1235	11.6	1134	1.818	978	66	168	237	318	39.98
10	8-25-32	8020	389	1003	1529	69394	399	445	1204	4088	1511	1.758	73482	103	1332	32.8	1214	9.9	1122	1.631	970	62	161	224	301	40.50
11	8-26-32	5963	390	1003	1529	52188	410	447	1204	3176	1510	1.756	55364	76.9	1297	25.0	1191	8.4	1110	1.428	962	58	155	208	279	40.86
12	8-27-32	3991	390	1002	1529	36182	408	447	1204	2098	1511	1.757	38280	51.4	1257	17.3	1163	7.0	1099	1.201	954	55	145	188	252	41.22
13	8-30-32	9953	379	905	1475	92858	395	439	1204	4945	1461	1.724	97803	134	1327	41.9	1210	11.8	1111	1.874	959	67	169	239	321	39.12
14	8-31-32	8024	380	900	1473	75815	404	445	1204	4027	1459	1.723	79842	110	1303	34.3	1192	10.4	1103	1.709	954	64	163	227	306	39.29
15	9-1-32	5971	392	902	1474	56844	407	445	1204	3082	1460	1.720	59926	81.4	1269	25.9	1167	9.0	1093	1.474	944	59	156	210	283	39.93
16	9-2-32	3988	390	902	1474	39034	416	448	1204	2105	1460	1.721	41139	56.1	1228	17.8	1139	7.7	1082	1.232	936	53	149	190	256	40.43
17	9-7-32	9958	381	800	1418	99610	397	443	1204	5085	1408	1.683	104695	138	1287	42.6	1180	12.4	1090	1.727	932	64	172	240	324	38.56
18	9-8-32	8014	389	800	1418	80707	403	446	1204	3865	1408	1.681	84572	110	1263	34.1	1159	10.3	1077	1.507	923	60	163	226	306	39.20
19	9-9-32	5982	390	801	1419	60197	408	447	1204	3026	1409	1.681	63223	82.5	1235	25.4	1139	8.6	1063	1.245	914	54	154	209	284	39.82
20	9-10-32	3978	388	801	1419	41551	414	448	1204	2087	1409	1.682	43638	56.6	1202	20.5	1124	7.3	1054	1.099	908	49	147	197	258	39.97
21	9-14-32	10000	386	698	1362	109406	399	445	1204	5570	1354	1.636	114976	146	1248	44.9	1148	12.5	1060	1.854	909	67	172	243	328	37.72
22	9-15-32	8009	388	698	1362	87291	411	447	1204	4168	1355	1.636	91459	115	1227	35.5	1132	9.9	1044	1.562	901	61	161	229	309	38.34
23	9-16-32	6009	388	697	1361	66082	417	448	1204	3061	1354	1.636	69143	84.9	1200	26.3	1109	7.5	1026	1.318	893	55	146	211	286	38.76
24	9-17-32	4011	387	699	1362	45307	423	451	1204	2129	1355	1.636	47436	59.8	1170	18.6	1085	5.2	1004	1.095	884	49	131	192	262	39.34

Ordinarily a turbine has but one supply of steam; that is, the amount coming through the throttle valve; but in this high-temperature unit the turbine has two sources of supply. The chief one is that of the throttle, and the secondary one is the steam supplied to the regulator to seal the high-pressure packing glands from which some of the sealing steam passes to the first-stage shell and thus through the remainder of the turbine. This secondary supply is used because steam of a lower temperature than the throttle steam is more satisfactory as sealing steam. This secondary supply of steam complicates to a considerable extent the calculation of the efficiency of the corresponding ideal unit, which is considered to fulfill the following conditions:

(1) The steam supplied to the ideal unit corresponds with that of the actual unit as to the quantity and condition at each of the two points where steam is admitted.

(2) All steam admitted to the ideal unit is assumed to have no throttling, leakage, fluid friction, turbulence, or heat transfer in its passage through the turbine. Any steam admitted, in addition to that at the throttle, is considered as mixed at con-

TABLE III

TEST DATA OF THE 10,000-KW HIGH-TEMPERATURE UNIT

Based on Actual Conditions of Steam at Throttle and at Turbine Exhaust

Run No.	Duration of Run, hr	Load, kw (L)	Throttle Steam: Press., lb per sq in. abs (p_a)	Throttle Steam: Temp., F (t_a)	Throttle Steam: Flow, lb per hr (w_a)	Sealing Steam: Condition at Entrance to Regulator: Press., lb per sq in. abs (p_b)	Condition at Entrance to Regulator: Temp., F (t_b)	Sealing Steam: Quantity, lb per hr: To 1st Stage Shell	Quantity: To 9th Stage Heater	Quantity: To 17th Stage Heater and Low-Press. Shaft Packing	Quantity: To Atmospheric Vent at High-Press. Shaft Packing	Quantity: Total (w_b)	Exhaust Press., In. Hg abs (p_5)	Temp. of Feed-Water Leaving 9th Stage Heater, F (t_f)	Enthalpy, Btu per lb: Steam Entering Throttle (h_a)	Enthalpy: Sealing Steam Entering Regulator (h_b)	Enthalpy: Saturated Liquid Corresponding to Feed-water Temp. (t_f) (h_f)	Energy Consumption of Unit, Btu per kwhr (E_e)	Thermal Efficiency of Unit, per cent (e_a)	Efficiency of Ideal Regenerative Unit, per cent (From Table II) (e_i)	Engine Efficiency of Complete Unit, per cent $\frac{(e_a)}{(e_i)}$	Generator Efficiency, per cent (From B.T.H.Co.) (e_g)	Engine Efficiency of the Turbine and Heaters, per cent $\frac{(e_a)}{(e_i e_g)}$
										Regulated Exhaust Pressure of Approximately 1.0 in. Hg abs													
1	3.997	10068	392	1003	84448	391	445	540	696	4090	118	5444	1.012	337	1529	1206	308	10731	31.81	41.86	75.99	95.58	79.30
2	4.999	8034	390	1006	67512	405	448	634	564	2970	88	4256	0.997	322	1530	1206	292	10890	31.34	42.00	74.62	95.07	78.49
3	6.722	5986	393	1004	50996	415	450	527	465	2074	64	3130	0.938	304	1529	1205	273	11186	30.51	42.28	72.16	94.11	76.68
4	9.995	4042	387	1005	36237	420	452	525	350	1346	38	2259	1.059	282	1530	1206	251	11995	28.45	41.59	68.41	91.90	74.44
5	[illegible]	10063	389	700	106691	395	452	384	977	4301	118	5780	0.975	342	1363	1211	313	11647	29.30	39.98	73.32	95.58	76.71
6	4.992	8045	388	703	85088	414	456	436	707	2828	88	4059	0.957	327	1364	1210	297	11747	29.05	39.90	72.81	95.04	76.61
7	6.687	6036	389	701	64587	425	459	242	576	1974	64	2856	0.995	310	1363	1211	278	12051	28.32	39.68	71.37	94.13	75.82
8	10.006	4038	390	703	44602	428	459	330	437	1311	38	2116	0.986	287	1364	1211	256	12739	26.79	39.78	67.35	91.88	73.30
										Unregulated Exhaust Pressure													
9	4.000	9960	389	1003	86030	396	445	1119	523	3218	118	4978	1.818	340	1529	1205	311	10968	31.12	39.98	77.84	95.55	81.47
10	5.000	8020	389	1003	69394	399	445	1171	482	2347	88	4088	1.631	326	1529	1204	296	11132	30.66	40.50	75.70	95.03	79.66
11	7.000	5963	390	1003	52188	410	447	1096	394	1622	64	3176	1.428	306	1529	1204	276	11461	29.78	40.86	72.88	94.08	77.47
12	10.000	3991	390	1002	36182	408	447	753	287	1020	38	2098	1.201	283	1529	1204	252	12078	28.26	41.22	68.56	92.23	74.34
13	4.019	9953	379	905	92858	395	439	1054	615	3158	118	4945	1.874	343	1475	1204	314	11274	30.27	39.12	77.38	95.55	80.98
14	4.965	8024	380	900	75815	404	445	1213	511	2215	88	4027	1.709	328	1473	1204	299	11547	29.56	39.29	75.24	95.04	79.17
15	6.713	5971	392	902	56844	407	445	1057	403	1558	64	3082	1.474	308	1474	1204	277	11873	28.75	39.93	72.00	94.08	76.53
16	10.061	3988	390	902	39034	416	448	791	310	966	38	2105	1.232	264	1474	1204	253	12453	27.41	40.43	67.80	91.80	73.86
17	4.017	9958	381	800	99610	397	443	1426	686	2855	118	5085	1.727	343	1418	1204	314	11498	29.68	38.56	76.97	95.55	80.55
18	4.986	8014	389	800	80707	403	446	1202	558	3017	88	3865	1.507	327	1418	1204	297	11727	29.10	39.20	74.23	95.02	78.12
19	6.700	5982	390	801	60197	408	447	1046	436	1480	64	3026	1.245	309	1419	1204	279	11940	28.58	39.82	71.77	94.10	76.27
20	10.085	3978	388	801	41551	414	448	774	338	937	38	2087	1.099	285	1419	1204	254	12667	26.94	39.97	67.40	91.78	73.44
21	4.000	10000	386	698	109406	399	445	1639	761	3052	118	5570	1.854	348	1362	1204	317	11927	28.62	37.72	75.87	95.56	79.40
22	4.999	8009	388	698	87291	411	447	1307	570	2203	88	4168	1.562	329	1362	1204	300	12045	28.34	38.34	73.92	95.02	77.79
23	6.670	6009	388	697	66082	417	448	973	450	1574	64	3061	1.318	310	1361	1204	280	12358	27.62	38.76	[illegible]	94.12	75.71
24	10.001	4011	387	699	45307	423	451	712	341	1038	38	2129	1.095	286	1362	1204	256	12996	26.26	39.34	66.75	91.63	72.69

stant pressure before entering the turbine with that at the throttle.

(3) The same number of ideal heaters are involved as in the actual unit; and the bleeding pressures are the same as those in the actual bleeder nozzles.

(4) The amount of steam bled to each ideal heater is calculated to equal that which is just sufficient to heat, at constant pressure and without any losses, all of the feedwater entering the heater to the saturation temperature corresponding to the pressure in the actual bleeder nozzle.

(5) The ideal heaters are considered to be of the "contact" type, and the water from each one, except the last, is considered as pumped without losses of any kind to the next one of higher pressure; from the last heater, the water is considered as delivered to the feed pump.

(6) The exhaust pressure is the same as that of the actual turbine.

Notation and Methods of Making Calculations

The notation that follows is used in Tables II and III and in the equations given in this section. In making the calculations for the ideal cycle, there are many states of the working substance involved as shown by those marked by the numerals in Fig. 5. On the other hand, for the actual unit only the states of the steam at a and b and that of the feedwater at f are needed; hence this figures does not show the probable states of the steam during its passage through the actual turbine.

Let w_a = actual flow of throttle steam, lb. per hr.;

w_b = actual flow of sealing steam, lb. per hr.;

$w_1 = w_a + w_b$ = actual total flow, lb. per hr.;

h_a = enthalpy, or heat content, of the steam at entrance to throttle, B.t.u. per lb.;

h_b = enthalpy of the sealing steam at entrance to the regulator, B.t.u. per lb.;

$h_1 = \dfrac{(w_a h_a + w_b h_b)}{w_1}$ = enthalpy of steam entering the corresponding ideal unit at state 1, B.t.u. per lb.;

p_a and p_b = steam pressures at entrance to the throttle and to the regulator, respectively, lb. per sq. in. abs.;

p_2, p_3, p_4 and p_5 = steam pressures at the 9th, 14th, and 17th-stage bleeder nozzles, and at entrance to the condenser, respectively, lb. per sq. in. abs.;

h_2, h_3, h_4 and h_5 = enthalpies of the steam after isentropic expansion from state 1 to p_2, p_3, p_4, and p_5, respectively, B.t.u. per lb.;

h_6, h_7, h_8 and h_9 = enthalpies of saturated liquid corresponding to p_5, p_4, p_3, and p_2, respectively, B.t.u. per lb.;

h_f = enthalpy of saturated liquid corresponding to actual feedwater temperature leaving the 9th-stage heater, B.t.u. per lb.;

L = load on the generator, kw.;

s_1 = entropy of the steam at pressure p_a after mixing that from the throttle with that from the regulator;

$m_2 = \dfrac{h_9 - h_8}{h_2 - h_8}$ = fraction of w_1 bled at state 2 in the ideal unit;

$m_3 = \dfrac{(h_8 - h_7)(1 - m_2)}{h_3 - h_7}$ = fraction of w_1 bled at state 3 in the ideal unit;

$m_4 = \dfrac{(h_7 - h_6)(1 - m_2 - m_3)}{h_4 - h_6}$ = fraction of w_1 bled at state 4 in the ideal unit;

$Wk_i = h_1 - h_5 - m_2(h_2 - h_5) - m_3(h_3 - h_5) - m_4(h_4 - h_5)$ = the mechanical energy available from the flow of 1 lb. of steam through the ideal regenerative unit, assuming this steam to enter the unit at state 1, B.t.u. per lb. of total flow;

$h_1 - h_9$ = net energy supplied the ideal unit with each pound of steam combined from both sources, B.t.u. per lb. of total flow;

$e_i = \dfrac{Wk_i}{h_1 - h_9}$ = efficiency of the ideal unit;

$E_c = \dfrac{w_1(h_1 - h_f)}{L} = \dfrac{w_a(h_a - h_f) + w_b(h_b - h_f)}{L}$ = energy consumption rate of the actual unit, B.t.u. per kw. hr.;

$e_a = \dfrac{3413}{E_c}$ = thermal efficiency of the actual unit;

$\dfrac{e_a}{e_1}$ = engine efficiency of the complete unit (turbine-generator and heaters);

e_g = efficiency of the generator, including bearings, ventilation and excitation.

The generator efficiencies, as given in Tables I and III, were supplied by the builder. The engine efficiency of the turbine and heaters is found by dividing the engine efficiency of the complete unit by the generator efficiency.

Sample calculations for the ideal and actual units in Run No. 1 (see Tables I and II) are as follows:

$$h_1 = \frac{w_a h_a + w_b h_b}{w_1 \text{ or } (w_a + w_b)} = \frac{84{,}448 \times 1529 + 5444 \times 1206}{89{,}892}$$
$= 1509$ B.t.u. per lb.

$$m_2 = \frac{h_9 - h_8}{h_2 - h_8} = \frac{317 - 237}{1355 - 237} = 7.16 \text{ per cent.}$$

$$m_3 = \frac{(h_8 - h_7)(1 - m_2)}{h_3 - h_7} = \frac{(237 - 163)(0.9284)}{1238 - 163}$$
$= 6.37$ per cent.

$$m_4 = \frac{(h_7 - h_6)(1 - m_2 - m_3)}{h_4 - h_6} = \frac{(163 - 48)(0.8647)}{1126 - 48}$$
$= 9.22$ per cent.

$$Wk_i = h_1 - h_5 - m_2(h_2 - h_5) - m_3(h_3 - h_5) - m_4(h_4 - h_5)$$
$$= 1509 - 945 - (0.0716)(1355 - 945) - (0.0637)(1238 - 945) - (0.0922)(1126 - 945)$$
$= 499$ B.t.u. per lb.

$$e_i = \frac{Wk_i}{h_1 - h_9} = \frac{499}{1509 - 317} = 41.86 \text{ per cent.}$$

$$E_c = \frac{w_1(h_1 - h_f)}{L} = \frac{89892(1509 - 308)}{10068} = 10{,}730 \text{ B.t.u. per}$$
kw. hr. = energy consumption rate of the actual unit.

$e_a = \dfrac{3413}{E_c} = \dfrac{3413}{10730} = 31.81$ per cent = actual thermal efficiency of the complete unit.

$\dfrac{e_a}{e_i} = \dfrac{31.81}{41.86} = 75.99$ per cent = engine efficiency of the complete unit.

$\dfrac{(e_a)}{(e_i)} \times \dfrac{(1)}{(e_g)} = \dfrac{0.7599}{0.9558} = 79.50$ per cent = engine efficiency of the turbine and heaters alone.

Energy Consumption Rate

It may be noted that throughout this paper the term "energy consumption rate" (energy rate) of the unit is used instead of the more common one "heat consumption rate" (heat rate). The former term is preferred because it is a more accurate expression for any steam turbine, although the term "heat rate" is entirely appropriate for a steam station. The reason for this distinction becomes apparent after careful consideration is given to the facts in the matter, regardless of what may have been heretofore commonly used. Steam passes through a turbine under steady flow conditions, and this steam cannot,

TABLE III-A

TEST DATA OF THE 10,000 KW HIGH-TEMPERATURE UNIT

Corrected to Throttle Pressure of 390 lb per sq in. abs, Exhaust Pressure of 1 in. Hg abs, and Nominal Throttle Temperatures as Tabulated

Run No.	Load, kw	Temp. of Steam at Throttle, F		Actual Pressure		Correction Factors				Throttle Flow lb per hr		Total Flow of Sealing Steam, lb per hr	Enthalpy, Btu per lb			Energy Consumption for Nominal Conditions Btu per kwhr
		Nominal	Actual	At Throttle, lb per sq in. abs	At Exhaust, in. Hg abs	For Throttle Temp.	For Throttle Press.	For Exhaust Press.	Total, $c_1 \times c_2 \times c_3$	Actual	Corrected to Nominal Conditions $(c) \times (w_a)$		Nominal Throttle Steam	Sealing Steam	Saturated Liquid Corresponding to Feedwater Temp.	
	(L)	(t_o)	(t_a)	(p_a)	(p_b)	(c_1)	(c_2)	(c_3)	(c)	(w_a)	$(w_{a'})$	(w_b)	$(h_{a'})$	(h_b)	(h_f)	$(E_{c'})$
Regulated Exhaust Pressure of Approximately 1 in. Hg abs																
1	10068	1000	1003	392	1.012	1.0001	1.0006	0.9999	1.0006	84448	84500	5440	1527	1206	308	10720
2	8034	1000	1006	390	0.997	1.0022	1.0001	1.0001	1.0024	67512	67670	4260	1527	1206	292	10890
3	5986	1000	1004	393	0.938	1.0005	1.0008	1.0036	1.0049	50996	51250	3130	1527	1205	273	11230
4	4042	1000	1005	387	1.059	1.0021	0.9995	0.9963	0.9979	36237	36160	2260	1527	1206	251	11940
5	10063	700	700	389	0.975	1.0000	0.9998	1.0006	1.0005	106691	106740	5780	1363	1211	313	11650
6	8045	700	703	388	0.957	1.0018	0.9998	1.0019	1.0035	85088	85390	4060	1363	1210	297	11770
7	6036	700	701	389	0.995	1.0002	0.9999	1.0001	1.0002	64587	64600	2860	1363	1211	278	12050
8	4038	700	703	390	0.986	1.0018	1.0000	1.0010	1.0028	44602	44730	2120	1363	1211	256	12760
Unregulated Exhaust Pressure																
9	9960	1000	1003	389	1.818	1.0020	0.9999	0.9727	0.9745	86030	83840	4980	1527	1205	311	10690
10	8020	1000	1003	389	1.631	1.0020	0.9999	0.9743	0.9762	69394	67740	4090	1527	1204	296	10860
11	5963	1000	1003	390	1.428	1.0020	1.0002	0.9792	0.9813	52189	51210	3180	1527	1204	276	11230
12	3991	1000	1002	390	1.201	1.0010	1.0001	0.9878	0.9889	36182	35780	2100	1527	1204	252	11920
13	9953	900	905	379	1.874	1.0033	0.9978	0.9710	0.9721	92858	90270	4950	1472	1204	314	10940
14	8024	900	900	380	1.709	1.0000	0.9978	0.9716	0.9695	75815	73500	4030	1472	1204	299	11190
15	5971	900	902	392	1.474	1.0014	1.0004	0.9771	0.9789	56844	55640	3080	1472	1204	277	11620
16	3988	900	902	390	1.232	1.0014	1.0001	0.9860	0.9875	39034	38550	2100	1472	1204	253	12280
17	9958	800	800	381	1.727	1.0000	0.9980	0.9756	0.9736	99610	96980	5090	1418	1204	314	11210
18	8014	800	800	389	1.507	1.0000	1.0000	0.9786	0.9786	80707	78980	3860	1418	1204	297	11490
19	5982	800	801	390	1.245	1.0009	1.0000	0.9870	0.9878	60197	59470	3030	1418	1204	279	11790
20	3978	800	801	388	1.099	1.0009	0.9997	0.9938	0.9944	41551	41320	2090	1418	1204	254	12600
21	10000	700	698	386	1.854	0.9988	0.9993	0.9715	0.9697	109406	106090	5570	1363	1204	317	11590
22	8009	700	698	388	1.562	0.9984	0.9998	0.9768	0.9748	87291	85090	4170	1363	1204	300	11770
23	6009	700	697	383	1.318	0.9983	0.9997	0.9826	0.9806	66082	64800	3060	1363	1204	280	12150
24	4011	700	699	387	1.095	0.9999	0.9994	0.9940	0.9933	45307	45000	2130	1363	1204	256	12910

therefore, continuously operate a turbine by merely having heat supplied to the fluid. Instead, the fluid must be constantly supplied with the more expensive mechanical energy delivered by suitable pumps before the absorption of heat takes place in a steam-generating unit. These two forms of energy supplied to the fluid are simply and accurately taken care of, in the equations previously given, by means of the enthalpies of the fluid at entrance to, and exit from the unit. In this connection, particular attention is called to the term h_f because this is the enthalpy of saturated liquid corresponding to the feedwater temperature leaving the ninth-stage heater, and not the enthalpy of the feedwater corresponding to its actual state leaving this heater. The actual pressure of the feedwater leaving this heater is several hundred pounds higher than the saturation pressure corresponding to the feedwater temperature, merely because the feed pump is conveniently connected between the ninth and fourteenth-stage heaters, as shown in Fig. 2. The three heaters and their pumps form part of the complete unit, but the feed pump is not included and thus the energy credited to the unit by the feedwater leaving the ninth-stage heater is determined by its temperature rather than by its temperature and pressure.

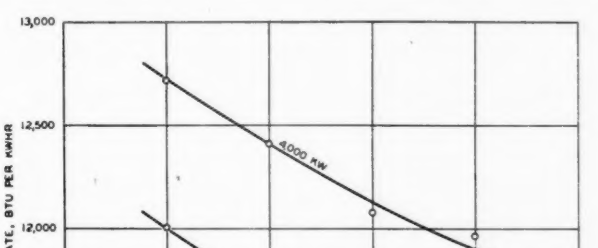

Fig. 6—Energy consumption rate of turbine-generator and three heaters

(Throttle pressure, 390 lb. per sq. in. abs.; exhaust pressure, 1.0 in. Hg abs.; and saturated sealing steam at throttle pressure)

Test Data

In addition to the summary, given in Table I, the principal test data are presented in Tables II and III for which the notation and methods of calculation have been given in the preceding section. The values given in Table II are needed to calculate the efficiency of the ideal turbine and three heaters under various conditions. In this table the pressures, temperatures, and rates of flow were obtained from observed values for the actual unit; and the enthalpies and entropies corresponding to the test data and the methods of calculation previously given were taken from Keenan's Steam Tables. In both Tables II and III, the data for the first eight runs were obtained with the exhaust pressure held as near to 1.0 in. Hg as could be obtained by regulating the air leakage into the suction line of the vacuum pump, since they were made when cold condensing water was available. The remainder of the runs, 9 to 24 inclusive, were made during warm weather when an exhaust pressure of 1 in. could not be obtained under full load conditions. For all of these runs, whether regulated or unregulated exhaust pressures, the engine efficiencies given in Table III have been based on the ideal unit having the same conditions as those prevailing during the test; hence this table gives the actual engine efficiencies obtained with various exhaust pressures and throttle temperatures. From this table one may observe that the highest value of the engine efficiency of the turbine and heaters for 1000-deg. steam and an exhaust pressure of 1.01 in. Hg abs. was 79.5 per cent, as shown for Run No. 1. On the other hand, in Run No. 9, when the exhaust pressure was 1.82 in. Hg abs. and the throttle temperature the same as in Run No. 1, the engine efficiency was 81.5 per cent. This is in keeping with the general characteristics of turbines, since the lower exhaust pressure involves a larger loss of available energy due to the greater losses due to moisture and to exit velocities from the wheels in the low-pressure region. The thermal efficiency in Run No. 1, however, is

greater than that in Run No. 9, because the available energy is so much greater with the lower exhaust pressure.

To show the relation between the throttle temperature and the rate of energy consumption of the unit, the data must be reduced to a comparable basis, as shown in Table III-A. The correction factors in this table were obtained from the builder of the turbine; and these corrections are seen to be almost negligible, except in certain runs where the exhaust pressure was much above the nominal value. Since the corrections due to variation in the exhaust pressure were appreciable for some runs, the manufacturer's values of this correction were checked by actual test and found to be correct. After obtaining the corrected energy consumption, as given in Table III-A, the results were plotted as shown in Fig. 6, in which the triangles represent data for Runs 1 to 8 inclusive, and the circles for Runs 9 to 24, inclusive. Even though the two sets of runs were made several months apart, as shown by Table II, the results agree extremely well for the 1000-deg. steam; this is largely due to the high degree of accuracy obtained by weighing all of the water from the condenser. For the supplementary runs with steam at 900, 800 and 700 deg. the results are also seen to be in satisfactory agreement, even though the condensate from the hotwell and heaters was measured by a venturi meter. The largest deviation from the curve is seen to be 1.1 per cent, for the small load of 4000 kw. and a steam temperature of 700 deg. These curves show that for all loads from 4000 to 10,000 kw., the energy consumption rate of the unit was decreased about 7 per cent by changing the throttle steam temperature from 700 to 1000 deg.

The additional curves given in Fig. 7 show approximately the same slope as those in Fig. 6. Each curve in Fig. 7, however, is slightly below its corresponding one of Fig. 6, because those in Fig. 7 represent the results obtained by using superheated sealing steam (t = 700 deg.) instead of saturated. The difference in the energy consumption rate, however, is not large especially at high loads, as may be seen from the curves in Fig. 8; but these results clearly indicate that there is a slight thermal gain by using the superheated sealing steam. Such a result is to be expected because a portion of the sealing steam leaks into the first-stage shell and there mixes with steam of much higher temperature. The tests made with the superheated sealing steam are considered merely as supplementary ones, since the turbine was designed to use saturated sealing steam; hence the tabular data on which Fig. 7 is based has been omitted.

TABLE IV

RADIATION AND CONVECTION LOSSES
FROM THE TURBINE AND HEATERS WITH 1000-DEGREE STEAM
AND A LOAD OF 10,000 KW

	Radiation Btu per Hr	Convection Btu per Hr	Total Btu per Hr
High-Pressure Cylinder and Cross-Over	51500	40700	92200
Low-Pressure Cylinder	14500	6100	20600
9th-Stage Bleeder Line	2700	2800	5500
14th-Stage Bleeder Line	21500	7200	28700
17th-Stage Bleeder Line	17300	7800	25100
9th-Stage Heater	5400	1600	7000
14th-Stage Heater	3600	1000	4600
17th-Stage Heater	2000	700	2700
Condensate Line from the 17th to the 14th-Stage Heaters, the Lines and Pump for the Heater Drains	4700	1700	6400
Condensate Line from the 14th to the 9th-Stage Heaters Including Pump	9000	4000	13000
Drain Line from the 14th to the 17th-Stage Heaters	4500	2000	6500
Drain Line from the 9th to the 14th-Stage Heaters	7000	3000	10000
Line from the Sealing Steam Pressure Regulating Valve to the Shaft Packing	1600	1000	2600
Total Losses	145300	79600	224900

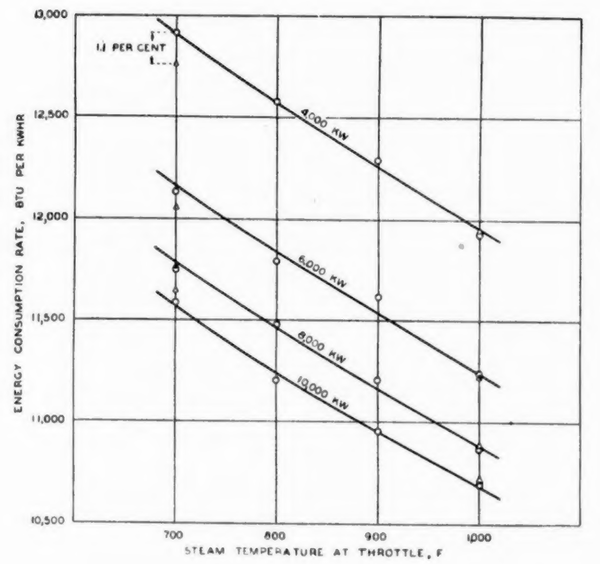

Fig. 7—Energy consumption rate of turbine-generator and three heaters

(Throttle pressure, 390 lb. per sq. in. abs.; exhaust pressure, 1.0 in. Hg abs.; and 700-deg. sealing steam at throttle pressure)

The engine efficiencies, as represented by the curves in Fig. 9, are plotted directly from Table III for Runs 1 to 8 with the regulated exhaust pressure.

These curves show that with 700-deg. steam the turbine would reach its maximum engine efficiency with a load only slightly above 10,000 kw., but that with 1000-deg. steam its maximum engine efficiency would be attained with an appreciably higher load—probably 13,000 or 14,000 kw.

The losses due to radiation and convection from the turbine and heaters are given in Table IV for a load of 10,000 kw. This table shows that radiation accounted for about 65 per cent of the total amount of heat transferred from the unit. This proportion decreased slightly for lower loads until it became 62 per cent with a load of 4000 kw. and 1000-deg. steam. For this load the total loss by heat transmission was calculated to be 192,600 B.t.u. per hr. In no case was this form of loss a serious one, because the unit was well insulated. The complete data for the observed temperatures and surfaces and the necessary calculations involved in the determination of these losses would require about 40 times as much space as that given in Table IV and could not, therefore, be easily given in this paper.

Energy Balance of the Turbine

An energy balance of a turbine is always of interest and value, because it shows at a glance the distribution of the net amount of energy supplied to the turbine. Such a balance has been prepared for Run No. 1, since this was the one for the largest load carried, and is for 1000-degree steam. The results given in Table V show that one-third of the energy supplied to the turbine was delivered by the turbine shaft, and two-thirds were absorbed by the condensing water, the radiation and convection losses being extremely small. After the energy absorbed by the condenser per pound of total flow is found, it then becomes possible, by the aid of the other data, to determine the state of the steam leaving the turbine. To make this calculation, the amount of steam passing to the condenser must be found in terms of the total flow to the turbine.

The weighed water from the hotwell was 71,280 lb. per hr. for Run No. 1; the condensate from the heater drains, as determined by an energy balance of the heaters, was 18,430 lb. per hr.; and the amount of steam escaping from the at-

TABLE V

ENERGY BALANCE OF THE TURBINE FOR A GENERATOR LOAD OF 10,068 KW AND A STEAM TEMPERATURE OF 1000 F

Energy Distribution	Btu per lb of Total Flow to Turbine	Per Cent
Energy delivered by turbine shaft . . . . From Table III, $\frac{3413 \times \text{Load}}{(\text{gen. eff.}) \times (w_a + w_b)} = \frac{3413 \times 10068}{0.9558 \times 89892}$	400.0	33.3
Energy lost by radiation and convection . . . . . $\frac{\text{Loss, from Table IV}}{\text{Total Flow}} = \frac{224{,}900}{89{,}892}$	2.5	0.2
Energy absorbed by condenser (to balance)	798.5	66.5
Energy supplied turbine . . . . From Table III, $\frac{w_a h_a + w_b h_b}{w_a + w_b} - h_f = 1509 - 308$	1201.0	100.0

mospheric vents of the shaft packing glands was 185 lb. per hr. Since the condenser received only 71,280 lb. of steam out of a total flow to the turbine of 89,892 lb., the amount of energy absorbed from each pound of steam entering the condenser is found, by the aid of Table V, to have been $\frac{(89{,}892)}{71{,}280}(798.5) =$ 1007 B.t.u. Then, since the condensate in the hotwell had a temperature of 78 deg. its enthalpy was 46 B.t.u. per lb., and the steam entering the condenser must have had a total amount of energy of 1007 + 46 or 1053 B.t.u. per lb. This total energy is made up of the enthalpy of the steam entering the condenser plus its velocity energy. Neglecting the velocity energy, temporarily, the enthalpy of the exhaust steam would then be 1053; and for the known exhaust pressure the corresponding moisture would be 4.0 per cent and the specific volume, 622 cu. ft. per lb., which may be used as a trial value to find the velocity. Since the area of the exhaust opening was 34.76 sq. ft. and the rate of flow to the condenser was 71,280 lb. per hr., or 19.8 lb. per sec., the exit velocity would be $19.8 \times 622/34.76 = 354$ ft. per sec., or 21,240 ft. per min. Such a result means that the velocity energy of the steam leaving the turbine would be 2.5 B.t.u. per lb.; and since the total energy of this steam has already been found to be 1053, its enthalpy would be 1053 − 2.5 = 1050.5 B.t.u. per lb. The corresponding value of the moisture then becomes 4.2 per cent; and the specific volume remains so close to the temporary value previously found that additional calculations of the velocity and moisture are not necessary.

Contrasted with this moisture of 4.2 per cent in the exhaust of the actual turbine, the steam from the corresponding ideal turbine would have 14.3 per cent moisture, as shown by state 5 in Fig. 5. This difference is due to the turbulence, which is caused by the steam in passing through the actual turbine, and which could be reduced to some extent by decreasing the exit velocity from each wheel. In other words, a larger number of stages or an increase in the diameters of the wheels would undoubtedly increase the efficiency of the unit. On the other hand, more or larger wheels mean a higher first cost because the turbine is made of expensive metals, and an economic balance of these opposing factors must be considered, even in an experimental machine, such as this one.

Loss Due to the Sealing Steam

The unusual arrangement of handling the sealing steam used for the high-pressure packing may possibly cause a difference of opinion as to the most logical method of calculating the loss of available mechanical energy produced by the leakage through the packing. Since about 90 per cent of the sealing steam is finally delivered to the seventeenth-stage heater, the question arises as to whether this heater pressure or the condenser pressure is the proper one on which the throttling loss of this part of the sealing steam should be based. If the condenser pressure be chosen as the proper one, the loss of available mechanical energy caused by using the saturated sealing steam at full load and 1000-deg. throttle steam amounts to 4.4 per cent of the total available energy; but if the seventeenth-stage heater pressure be chosen as the proper base, the loss is only 2.3 per cent. The authors consider the 4.4 per cent to be the more logical value for reasons that will now be given.

TABLE VI

FEEDWATER - HEATER PERFORMANCE DATA

Run No.	Turbine-Generator Load, kw	Actual Temp. of Steam at the Throttle, F (L)	Turbine Stage of Heater Connection (t_a)	Heating Surface sq ft	Steam from Extraction Nozzles and Shaft Packing: Flow, lb per hr	Steam from Extraction Nozzles and Shaft Packing: Press. of Steam at Entrance to Heater, lb per sq in. abs (p_e)	Steam from Extraction Nozzles and Shaft Packing: Temp. of Steam at Entrance to Heater, F (t_e)	Steam from Extraction Nozzles and Shaft Packing: Temp. of Saturated Steam Corresponding to (p_e), F ($t_{e'}$)	Temp. of Drains from Heater, F	Feedwater: Flow, lb per hr	Feedwater: Temp. at Entrance to Heater, F	Feedwater: Temp. at Outlet of Heater, F (t_o)	Feedwater: Temp. Rise in Heater, F	Terminal Temperature Difference: Actual, ($t_e - t_o$)	Terminal Temperature Difference: Nominal, ($t_{e'} - t_o$)	Mean Specific Heat of Water in Heater, Btu per lb per F	Heat Absorbed by Water in Heater, 1000 Btu per hr	Nominal Logarithmic Temp. Diff., Based on the Saturated Steam Temp. $t_{e'}$, F	Average Velocity of Water through Tubes, ft per sec	Nominal Overall Coefficient of Heat Transfer, Based on the Saturated Steam Temp. ($t_{e'}$) Btu per hr per sq ft per F
1	10068	1003	9	284	6219	122.6	717.0	342.9	----	89680	264.7	336.7	72.0	380.3	6.2	1.030	6651	14.9	7.83	1571
			14	303	5424	39.7	636.8	266.8	----	89680	191.7	260.9	69.2	275.9	5.9	1.009	6262	14.3	6.24	1445
			17	463	6756	10.2	362.8	194.1	193.6	71281	78.3	190.9	112.6	171.9	3.2	0.996	7994	19.1	3.16	905
2	8034	1006	9	284	4655	97.9	692.7	326.2	--	71595	253.5	321.8	68.3	370.9	4.4	1.027	5022	13.4	6.21	1320
			14	303	4226	31.9	519.6	253.7	--	71595	182.0	249.6	67.6	270.0	4.1	1.007	4874	13.1	4.99	1228
			17	463	5134	8.2	352.3	182.9	183.9	57580	75.4	181.4	106.0	170.9	1.5	0.997	6085	16.1	2.53	815
3	5986	1004	9	284	3278	72.8	662.2	305.6	--	53971	239.1	303.7	64.6	358.5	1.9	1.020	3556	11.0	4.66	1138
			14	303	3085	24.0	501.2	237.8	--	53971	169.6	235.2	65.6	266.0	2.6	1.005	3550	11.8	3.73	995
			17	463	3600	6.2	333.7	170.8	171.5	44008	72.3	169.6	97.3	164.1	1.2	0.995	4261	14.5	1.94	635
4	4042	1005	9	284	2083	52.4	632.6	284.0	--	38392	223.9	282.3	58.4	350.3	1.7	1.016	2278	10.0	3.29	802
			14	303	2086	17.0	482.2	219.4	--	38392	155.7	218.3	62.6	263.9	1.1	1.002	2408	9.8	2.66	810
			17	463	2266	4.3	308.1	154.9	155.9	31957	71.1	155.7	84.6	151.4	0.8	0.996	2693	--	1.41	—
5	10065	700	9	284	9189	138.0	495.8	350.2	--	112271	267.1	342.2	75.1	153.6	8.0	1.031	8693	16.2	9.86	1889
			14	303	6998	43.5	329.2	272.3	--	112271	197.6	263.3	65.7	65.9	9.0	1.011	7457	14.9	7.82	1651
			17	463	9258	11.6	200.3	200.3	200.1	86826	78.4	196.4	118.0	3.9	3.9	0.997	10215	20.5	3.86	1076
6	8045	703	9	284	6795	106.6	411.7	332.4	--	88985	256.0	327.4	71.4	144.3	5.0	1.029	6538	14.2	7.71	1621
			14	303	5483	35.0	312.9	259.3	--	88985	187.5	252.3	64.8	60.6	7.0	1.008	5812	14.0	6.20	1370
			17	463	6873	9.3	189.8	189.8	189.7	69834	77.2	186.5	109.3	3.3	3.3	0.997	7610	18.8	3.10	874
7	6036	701	9	284	4758	80.6	442.8	312.5	--	67317	242.4	309.5	67.1	133.3	3.0	1.022	4616	12.3	5.81	1321
			14	303	4016	26.1	293.8	242.4	--	67317	175.9	238.9	63.0	54.9	3.5	1.005	4262	12.0	4.67	1172
			17	463	4749	7.2	178.0	178.0	177.0	53794	77.0	175.4	98.4	2.6	2.6	0.996	5272	15.5	2.38	690
8	4038	703	9	284	2929	56.0	412.4	288.2	--	46626	225.9	286.6	60.7	125.8	1.6	1.017	2878	10.2	3.99	993
			14	303	2566	18.1	274.9	222.7	--	46626	160.9	219.4	58.5	55.5	3.3	1.002	2733	11.2	3.19	805
			17	463	2956	4.9	165.3	161.1	161.8	38175	73.7	160.2	86.5	5.1	0.9	0.996	3209	12.6	1.68	564

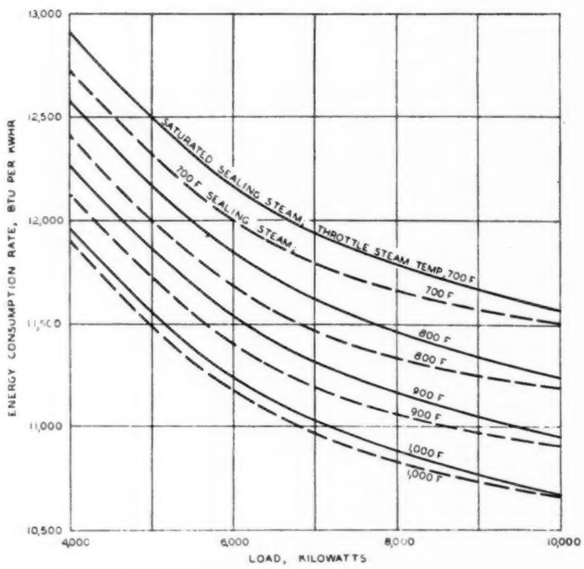

Fig. 8—Energy consumption rate of turbine-generator and three heaters

(Throttle pressure, 390 lb. per sq. in. abs.; exhaust pressure, 1.0 in. Hg abs.) Derived from Figs. 6 and 7

If there were no feedwater heaters used, the condenser pressure would clearly be the one to which all the steam supplied the turbine would be expected to expand before leaving the turbine. With a regenerative feedwater heating system, all the heat delivered to the heaters is supposed to come from steam that has delivered some mechanical energy to the turbine blades before extraction, and thus an extremely high thermal efficiency is obtained from such steam because the bled steam rejects no energy to the condenser. The larger the proportion of throttle steam that may be extracted from the turbine for use in the heaters, the higher will be the thermal efficiency of a regenerative unit. On the other hand, when the leak-off from the sealing steam is delivered to heat the regenerative feedwater heaters, the amount of steam that may be bled from the turbine is thereby reduced and thus the advantages of the regenerative system are taken away from a certain amount of throttle steam that must now pass entirely through the turbine to the condenser. Since that part of the sealing steam that passes to the heaters delivers no work to the turbine blades, the result is not thermodynamically equivalent to that which would be obtained with the same feedwater temperature derived entirely from bled steam. Consequently the loss of available mechanical energy, due to the portion of the sealing steam that flows from the packing to the heaters, is calculated as if this steam had been throttled to condenser pressure. About 10 per cent of the sealing steam leaked through the inner shaft packing to the first-stage shell, and for this portion the loss of available energy is simply that due to the throttling caused by passing through the regulator and the inner packing.

Heater Performance

The three four-pass feedwater heaters which received steam bled from the turbine form part of the complete unit, and the data relating to the performance of these heaters, for runs 1 to 8, inclusive, are given in Table VI. Possibly the most interesting information obtainable from the table is that pertaining to the terminal temperature differences and the rates of heat transfer in the heaters. The terminal temperature difference of a feedwater heater is commonly taken as the temperature of saturated steam corresponding to the pressure of the steam entering the heater minus the temperature of the feedwater leaving the heater; and this difference is called the "nominal" one in this table. The enormous differences between the actual and "nominal" values of the terminal temperature difference when superheated steam is used are clearly shown.

The last column of Table VI shows the "nominal" over-all coefficients of heat transfer for the three heaters for various conditions of operation. It should be noted that when highly superheated steam is bled to a heater, the "nominal" coefficient is very high, as, for example, in the ninth-stage heater in Run No. 1. In this case the steam entering the heater was superheated 374 deg. (717–343), and by the usual method of calculation, the logarithmic mean temperature difference is only 14.9 deg., whereas the actual mean temperature difference is much greater. No general method of calculating the real mean temperature difference where superheated steam and latent heat are both involved has been accepted, and thus the nominal values only are given. Regardless of what may appear from these nominal values, highly superheated steam does not increase the real coefficient of heat transfer, but does increase very much the difficulties of determining the actual mean temperature difference and also the actual coefficient.

Acknowledgments

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Future Possibilities and Probabilities

After examining the test data obtained from an experimental machine, such as this 1000-deg. steam turbine, engineers naturally desire to interpret such results in terms of the future possibilities and probabilities. However, in attempting to predict how good a performance may be expected from a large turbine using 1000-deg. steam, and considering the probability of such a turbine being built in the immediate future, one must consider a number of factors that are necessarily somewhat indeterminate at the present time. Thus one can-

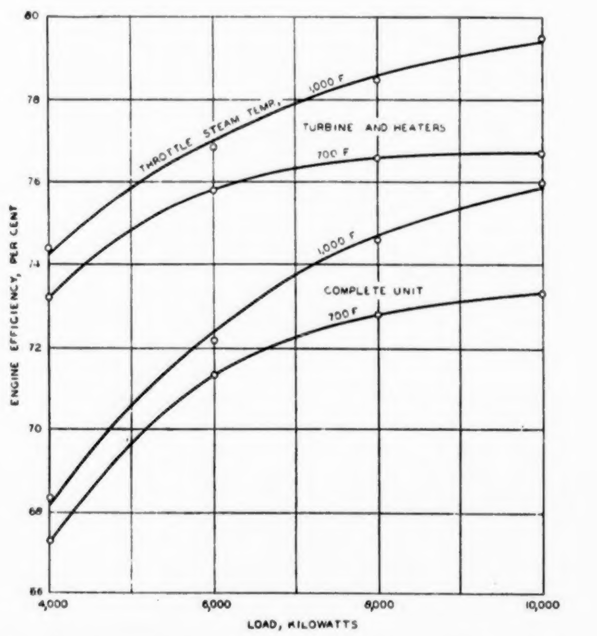

Fig. 9—Engine efficiency of turbine-generator and three heaters and of turbine and heaters alone

(Throttle pressure, regulated to approximately 390 lb. per sq. in. abs.; exhaust pressure, regulated to approximately 1.0 in. Hg abs. and saturated sealing steam at throttle pressure)

not be certain how successful future designers may be in reducing or eliminating the loss due to the leakage of steam past the high-pressure shaft packing; but with a large turbine, say 75,000 kw. or more, the possibility of making the high-pressure cylinder of the double-flow type seems attractive, as such a scheme would eliminate entirely this loss. Furthermore, the elimination of the high-pressure packing glands would counteract the extra length of turbine shaft required by the double-flow type, and thus the distance between the bearings of the high-pressure cylinder would not be materially different from that required in a single-flow type with long shaft-packing glands.

The energy consumption rate of a very large turbine as determined by that of a relatively small one, such as a 10,000-kw. machine, is also a matter about which there is certain to be a difference of opinion, because such comparisons are so often based on experimental data relating to turbines having many basic differences other than their sizes. The estimate made by the authors has already been given in the early part of the paper as about 9900 B.t.u. per kw. hr. for 50,000-kw. units when operating with 1000-deg. steam, 390 lb. per sq. in. abs., and 1 in. Hg abs. exhaust pressure. For a 100,000-kw. unit this rate might be brought down to 9600 or better; and for 1200 lb. per sq. in. without reheating one might expect close to 8600 B.t.u. per kw. hr. This would mean that the moisture in the last stage has reached or exceeded the allowable limit set by many turbine engineers, for it would be about 12 per cent. If reheating were employed with the high-pressure and 1000-deg. throttle steam, the energy consumption rate of the unit might then be brought down to about 8400 B.t.u. per kw. hr.

The estimates previously given are further strengthened by the evidence furnished by several turbine builders in this country, who estimate that a 10,000-kw. unit with three regenerative feedwater heaters and designed for 700-deg. steam at the throttle should have an energy consumption rate of about 10,860 B.t.u. per kw. hr., when the throttle pressure is 390 lb. per sq. in. abs. and the exhaust pressure is 1 in. Hg abs. By referring to Fig. 6 or 7, it may be seen that such a result is nearly 6 per cent better than that obtained from the Detroit turbine when operated with 700-deg. steam. This difference is likely due in a large measure to the large shaft-packing leakage and possibly to a large interstage leakage. Consequently, if a 10,000-kw. unit can be built for 1000-deg. steam and operated without serious leakage, the slope of the curves in Figs. 6 and 7 show that such a unit would have an energy consumption rate of about $(1.000 - 0.07) \times 10{,}860$ or 10,100 B.t.u. per kw. hr. Then for large units and high-pressure steam this rate may possibly be brought down to the figures previously given.

The cost of materials suitable for use with 1000-deg. steam will probably be materially reduced in the future, if one may judge from similar developments of various other metals. As the development progresses and the production is increased, the costs of these materials are almost certain to be greatly reduced. Then the progress in the use of high-temperature steam will probably be greatly accelerated.

New Bronzes on Market Restricted Output Now Extended

For many years The Superheater Company has been producing bronze castings at their East Chicago, Ind., plant, for their own use and for the use of local manufacturers. As a result of increased demands on the company for high-grade, uniform castings, plant and laboratory facilities have now been extended to provide a service on a much broader basis.

The company produces bronze castings, rough or finished, in three distinct classes of mixtures, namely, standard bronzes, aluminum bronzes and super-tensile manganese bronze.

Reports on High Temperature Creep and Fatigue

Investigations on the creep and fatigue of cast and wrought, high- and low-carbon 18-Cr, 8-Ni type steel under high temperature as carried on at Battelle Memorial Institute and the University of Illinois, under the sponsorship of a joint A.S.M.E. and A.S.T.M. committee are reported by H. C. Cross in a paper at the A.S.M.E. Annual Meeting.

The room-temperature tensile strengths of the wrought materials were roughly a quarter higher than those of the cast materials, and those of the high-carbon lot somewhat higher than those of the low-carbon. In both fatigue and creep tests, high-carbon wrought was at the head and low-carbon at the foot of the list in load-carrying ability. Carbon had a strengthening effect on creep and fatigue resistance of these steels, in line with published information.

The order of strength of the materials as shown in room-temperature tests carries through the high-temperature tests, wrought material being stronger than cast, though the creep resistance of the two is not very different at 1200 fahr. The generalization sometimes stated that coarse-grained cast material has superior creep resistance to fine-grained wrought material is not borne out in this case. Such comparisons should state the type of material and the temperatures for which they are made. In creep tests, the cast materials were characterized by higher initial deformation than the wrought. Some bars of cast high-carbon fractured in creep tests with very little elongation.

Contrary to the general opinion that all "unstabilized" 18-8 of carbon content over 0.02 per cent shows embrittlement as judged by room temperature impact tests after sojourn at 1100 to 1400 fahr. the low-carbon heat, cast or wrought, water-quenched or air-quenched, showed absolutely no reduction in impact resistance on test after being subjected to load at 1000 to 1200 fahr. for periods of the order of 750 to 1750 hr. No certain explanation for this lack of impact embrittlement is at hand. Much of the high-carbon cast and wrought material came through creep tests of similar periods and temperatures without notable loss of impact resistance.

Carbide precipitation occurred at elevated temperatures in both high and low carbon, and was no more marked in the high than in the low. Though internal changes occur, the creep curves show no irregularities to indicate coincident alteration in creep resistance. Creep curves for all four lots kept changing in slope during the earlier part of the creep tests, a fairly constant rate of creep not being approached till more than 500 hr., as a rough average, had elapsed. Many of the accelerated methods that have been suggested for determination of creep properties thus appear inapplicable to steels of this type.

Combustion Engineering Company, Inc. Announces New Appointments

James Cleary, who has been a sales executive of the company for many years, has been appointed manager of the Philadelphia sales district and will make his headquarters at the company's Philadelphia office, 1616 Walnut Street.

Fred L. Farrell has been appointed manager of the New England sales territory and will make his headquarters at the company's Boston office, Chamber of Commerce Building. Mr. Farrell has had a long identification with sales work in this field, and was previously associated with this company in the same office he now holds.

G. O. French has been engaged as a sales engineer specializing on sales work in connection with the company's fire-tube boilers and all special shop work such as oil refinery equipment, chemical equipment, special alloy vessels, tanks, plate work, etc. Mr. French was previously New York District Sales Manager of The Air Preheater Corporation, and prior to that a sales engineer of the Walsh & Weidner Boiler Co.

A New Boiler-Water Treatment for the United States Navy

By THORVALD A. SOLBERG[1]

and

By ROBERT C. ADAMS, Jr.[2]

U. S. Naval Engineering Experiment Station,
Annapolis, Maryland

Extensive studies were made on samples of feedwater and scale from twenty ships located on widely separated stations. This was followed by comparative trials of the colloidal, coating, electrolytic and chemical methods of feedwater treatment, using two 600-lb. boilers at the U. S. Naval Experiment Station. As a result of these investigations a new Navy formula was evolved, consisting of anhydrous disodium phosphate, soda ash and corn starch, in proportions of 47, 44 and 9 per cent, respectively.

FROM the time that steam propulsion was adopted by the United States Navy until less than twenty-five years ago there was no uniformity in the methods of boiler-water treatment in use. Little trouble was experienced from scale formation or carry-over because of the combination of low working pressure, low rate of evaporation and the use of distilled water for make-up feedwater. Such scale as did occur was considered to be unavoidable and was removed regularly by periodic cleaning. The use of a small amount of lime or soda to give the feedwater a slight alkalinity had the semblance of official sanction. The engineering personnel in many cases were permitted to choose, subject to slight restriction, the compound they had found to give the best results. This situation exists in some steamship companies today, although many operators have a standardized treatment.

First Study of Boiler-Water Treatment for the Navy

The records of the Navy show that during the first decade of the century the trouble from scale formation and carry-over increased, although the general use of fire-tube boilers at that time served to minimize the trouble from scale formation. Some action became necessary, however, as the troubles increased, so that in 1909 Lieutenant Commander Frank Lyon[3] was detailed to make a study of boiler-water treatment.

Between 1909 and 1911, Commander Lyon made a rather complete study of the various corrosion inhibitors which were available at that time. Most of his corrosion experiments were performed with steel specimens exposed in open beakers. Some experiments were performed with a small boiler from a steam launch, and there also was a test made of an electrolytic system. Other experiments were made in bombs made from boiler-tube sections.

The best and cheapest protection against corrosion found by Lyon was a three per cent normal solution of sodium carbonate. The maintenance of such an alkalinity in the boiler water was tried on some vessels of the Fleet. Almost immediately it was discovered that such high alkalinity induced severe carry-over from steaming boilers. A re-survey of Lyon's copious data showed that quite satisfactory protection against corrosion could be obtained with an alkalinity of only 0.1 to 0.5 per cent normal (2.9 to 14.6 gr. per gallon as calcium carbonate). The limits were reduced to these values, which have been maintained since that time.

The boiler compound which has been used for the maintenance of alkalinity is virtually the same as that devised by Lyon. It consisted of ten per cent of sodium phosphate, one per cent of potato starch, sufficient extract of tannin-bearing material to give two per cent of tannic acid, and the balance soda ash. These various ingredients were intended for different purposes. The soda ash was to provide the necessary alkalinity. The organic materials, tannin and starch, were expected to absorb oxygen from the feedwater and so reduce corrosion. It was believed that the sodium phosphate lowered the surface tension and thus minimized the possibility of carry-over, and also that it had some effect on scale prevention.

The proportions of this compound have been changed slightly during the past twenty years, but the compound has remained essentially as originally specified by Lyon in 1911. The phosphate has combated the scale-forming salts, and the soda ash, in maintaining the alkalinity, also has provided considerable carbonate for scale prevention. The combination has been so satisfactory for a wide variety of boiler waters that there has developed a wide market for "Navy powder," which is used on many merchant vessels, as well as in a host of inland power plants.

Because of the many advances made in the art of boiler-water conditioning during the past decade and the inability of this system to give the required results in modern marine boilers, the Bureau of Engineering, in 1931, authorized an investigation of boiler-water treatment at the U. S. Naval Engineering Experiment Station at Annapolis, Maryland. The object was to make a thorough study of the subject and, as a result, to devise a satisfactory system for use on all naval vessels. This investigation was completed in July 1933, and the subject matter of this paper deals, in a general way, with the conduct of that work.

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[2] Chemical Engineer, U. S. Naval Engineering Experiment Station.
[3] Now Captain, retired.

Development of the Investigation

A survey was made of the conditions existing in the Fleet as the first step in defining the work of the investigation. Eight samples of water from different places in the feedwater cycle were collected from each of twenty ships. The ships chosen were of various classes, located in widely separated stations. Analyses of these waters, either in full or in part, were made in order to determine the range of conditions to be met. Samples of scale from each of two boilers were obtained from the same vessels.

The numerous analyses which were made of these waters showed conclusively that definite, scale-forming conditions usually were present. In every case the boiler waters tested showed hardness values varying from three to fifteen grains per gallon. Scale analyses showed the presence of both sulphates and silica, with some intermixture of carbonate and the usual amounts of corrosion products. The scale analyses were not considered representative of the average scale known to exist because the method of sampling was deficient. The usual method of collecting a sample of scale aboard ship consisted of running a motor-driven brush through several tubes and catching the dirt obtained. Actual tests have shown that this gives only a surface layer which consists largely of sludge products and is not representative of the real scale beneath.

Similarly, no standard method of sampling the waters was used and consequently full reliance could not be placed on the results of the analyses made. Two very illuminating discoveries, however, were derived from these analyses. The first was that in most of the vessels in the Navy the boiler water alkalinity was maintained at or near the lower limit of 0.1 per cent normal (2.9 gr. per gallon). The other was the fact that a large part of the feedwater contamination, instead of being the result of condenser leakage or evaporator carry-over, accumulated while the water was stored in the reserve bottoms.

The method chosen for the solution of the Navy's needs in boiler-water treatment was to test the available commercial methods to determine their various advantages. Following the comparative tests, and based upon the results obtained from them, it was planned to make such further experiments and test runs as accumulated experience would indicate to be desirable.

A study of the analyses of scale and water samples from the Fleet, together with the data from preliminary runs in a boiler which had been assembled from scrap material, indicated the conditions desired in a comparative test. The time limit of the investigation made a short time test imperative. Within a maximum time of 200 hours' steaming it was desired to include the possibility of heavy scale deposit, severe corrosion, and unfavorable conditions of boiling and sludge deposition. All the conditions were to be similar to, but of greater intensity than those existing in the Navy.

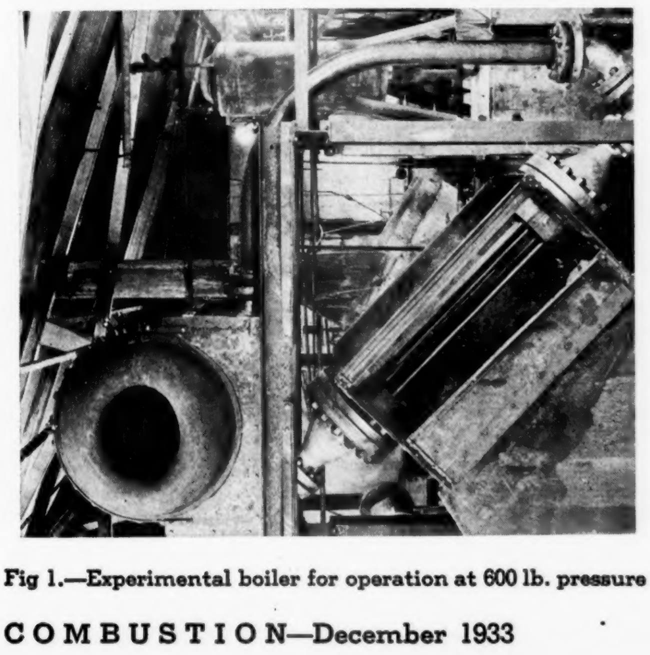

Fig 1.—Experimental boiler for operation at 600 lb. pressure

Two 600-lb. boilers were designed and built for this work. The design of the boiler was such that results of practical value could be obtained at the same time that control, approaching laboratory precision, could be maintained. A major consideration was the provision for ease in cleaning, overhaul and assembling, so that little time need be lost between successive test runs.

A photograph of the first unit constructed, with the side casing removed to show the general design features is shown in Fig. 1. Each boiler consisted of a single drum, 30-in. inside dia. and 4 ft. long. The seam at one end was riveted and that at the other end was fusion-welded. The drum was pierced near the bottom by two 4-in. openings for the upcomer and downcomer; and by four 2-in. openings in the upper half for the feed-line, the steam line to the feed pump, the safety valve and the steam line to the condenser.

The tube nest, which was set at a 45-deg. angle, was made up of fourteen 1-in. tubes, 51 in. long between the tube sheets. The tubes were arranged in four rows of alternately four and three tubes each. The total heating surface amounted to $16^2/_3$ sq. ft., but part of each end of the tube nest was protected from the flame by a cast-iron collar supporting firebrick. The effective heating surface was thus about 15 sq. ft.

The boiler had no lower drum nor any lower space in which sludge or solids could settle out of suspension. All solids in the boiler water were forced to remain in circulation. The circulation in all tubes was necessarily upward. This arrangement was intended to increase the susceptibility of the boiler to carry-over, but the comparatively great area of boiler-water surface and the large steam space combined to make carry-over most improbable.

Each boiler was fired with oil using air atomization. The furnace construction gave some troubles initially, principally from frequent burning out of baffles. In the final arrangement only one baffle was used, placed directly over the top row of tubes. Because of the wide tube spacing used, the comparatively large furnace volume, and the single-baffle construction, practically the entire surface of most tubes was exposed to radiant heat. This also was evidenced by the almost uniform layers of scale formed over the water side of the whole tube surface. The furnace temperatures and conditions obtained were fully as severe as those found in any large size boiler. The over-all rate of evaporation in the tests varied from fourteen to eighteen pounds of water per hour per square foot of evaporating surface which approximates conditions in modern naval boilers at full power.

The test procedure which was developed for use with the experimental boilers is here given in detail.

1. Initial tests were made at 350 lb. working steam pressure and if found satisfactory later tests were made at pressures up to 600 lb. The available time did not permit carrying out all desired 600-lb. tests.

2. The duration of a run was between 150 and 200 hours, which were continuous except for a break of 48 hours over a week-end. Usually a run was commenced on a Monday and concluded on Wednesday or Thursday of the second week. The boiler was opened for inspection on the following day (Thursday or Friday).

3. On commencement of the test the boiler was filled with water of about twice the concentration of the normal feedwater. The normal feedwater was contaminated by a standard solution of gypsum in Severn River water and by the separate addition of a small amount of sodium silicate. The feedwater thus contaminated had the following characteristics:

(*a*) A chloride content of 4–6 gr. per gallon. This was supplied by using water from the Severn River, which has all the salts found in sea water and in the same relative proportion, but in a concentration varying from one-fifth to one-half of that of sea water. In this way all the contaminating salts

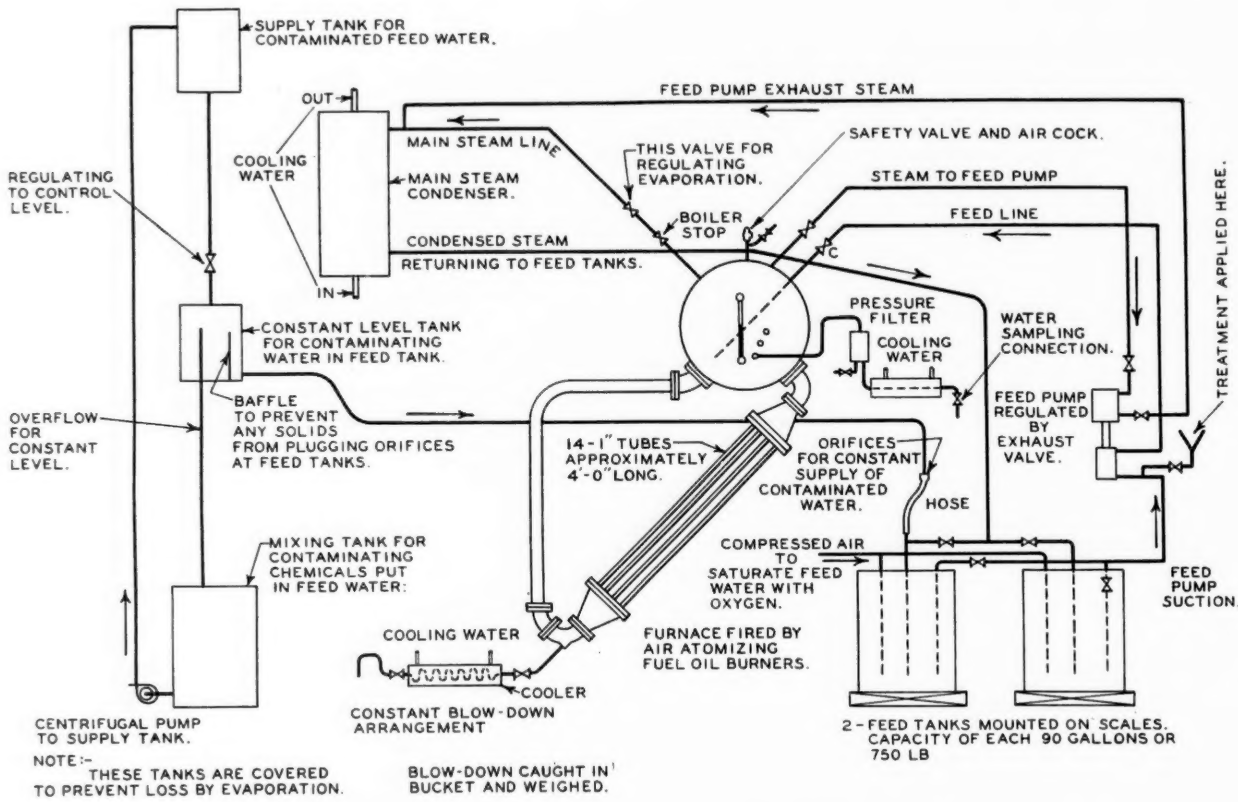

Fig. 2—Arrangement of piping and equipment for experimental boiler

found in sea water were fed to the boiler in a manner simulating condenser leakage.

(*b*) The total calcium sulphate in the feedwater was kept between 65 and 70 parts per million.

(*c*) The feedwater was kept saturated with air (oxygen).

(*d*) The silica content of the feedwater was maintained at 6 parts per million of SiO_2 by the addition of water glass.

4. When the chloride concentration in the boiler reached 60 gr. per gallon a continuous bottom blow was started and maintained at such a rate as to keep the chloride content below 70 gr. per gallon.

5. The rate of evaporation in the boilers was between 14 and 18 lb. per sq. ft. of heating surface—average total 220 lb. per hour per boiler.

6. Corrosion specimens of Class B steel, about $1^1/_2 \times 3 \times {}^1/_8$ in., were mounted totally submerged in the steam drum in order to obtain data on the relative amount of corrosion.

7. Compound or treating chemical was added intermittently, usually every four hours, direct to the feed-pump suction.

The contaminating solution was stored in a 270-gal. tank from which it could be pumped to an overhead tank as necessary. From the overhead tank it flowed by gravity to a constant-level tank, from which it was distributed through orifices to each of the boilers. A line diagram of the entire installation is shown in Fig. 2. Two weighing tanks on platform scales were provided for the feedwater and condensate. The steam generated by the boiler was returned as condensate to the same tank into which the contaminating solution flowed. When the condensate tank was full, both tanks were weighed, and the flows were reversed so that the condensate tank became the feed tank and vice versa. The rate of flow from the orifice was known so that the weight of the contaminating solution could be subtracted for the calculation of the rate of evaporation.

The orifice was a hole drilled in the cap at the end of the delivery line. It was of the proper size to deliver one and $1^1/_8$ gal. per hour. The water delivered through this orifice mixed with the condensate to give a feedwater which had 5 gr. per gallon of chloride, as Severn River water, plus sixty-five parts per million of calcium sulphate, as gypsum. The Severn River water contained some additional sulphate so that the ratio of sulphate to chloride in the feedwater was about 0.70. This ratio was approximately the same as that found in the samples of make-up feedwater which had been collected from the Fleet. Sufficient water glass was added to the new feedtank at each change-over to give six parts per million of silica to the feedwater. Although silica contamination occurs in the Service, the silica could not be added to the concentrated contaminating solution because of the low solubility of the heavy metal silicates, but it could be added separately to the feedwater without causing precipitation.

The disposition of the corrosion specimens in the boiler drum is shown in the isometric sketch of Fig. 3. Care was taken to insure good electrical contact between the specimens and the rack, and between the rack and the boiler shell.

Two regular analyses of the boiler water were made daily, and extra analyses were made as needed. These analyses comprised determinations of carbonate, hydroxide, total alkalinity, sulphate, chloride, soap hardness and pH. Carbonate, hydroxide and total alkalinity were determined by the strontium-chloride method of Schroeder.[4] Originally all sulphates were run gravimetrically, but, after tests showed its accuracy to be satisfactory, the benzidene method was used because of its greater speed. Chloride was titrated with silver nitrate, and pH was determined colorimetrically. Special analyses, which were required for some of the methods of treatment, also were run twice daily.

A reference run, using the test procedure outlined above, but without any treatment, was made with each boiler before beginning the regular series. This blank run gave excellent information on the behavior of the water and on the operation of the boiler system. It demonstrated that a hard, adherent scale, about 0.035 in. thick, could be formed in 175 hours of

[4] Schroeder, W. C., and Fellows, C. H., Progress Report of Sub-Committee No. 8, Joint Research Committee for Boiler Feedwater Studies, A.S.M.E. Annual Meeting, 1931.

TABLE I—SUMMARY OF DATA COLLECTED IN A TEST RUN

Date	Time	Hours' Evaporation	pH	CO_2	OH	Chloride P.p.m.	Chloride Gpg.	SO_4	PO_4	SiO_2	Hardness	% N.A.	SO_4/Cl	A.S.M.E. Ratio	Lb./hr. Evap.	Av. Press.	Lb./hr. B. Blow	Grams Compound
June 19	1600–1600	0–6	Added 300 lb. Ca-Sr Solution.					Started orifice running.							294	310		635D71S
June 19	1500	5	12.0	84	85	392	22.9	385	500	...	0	1.00	1.00	1.07				
June 19 & 20	1600–0800	6–22													235	305		300
June 20	0620	20.6	12+	144	158	718	42.0	886	770	...	0	1.41	1.23	1.75				
	0800–1600	22–26						Added 180 lb. Ca-Sr Solution							239	307	2.0	100
	1610	26.2	12+	132	124	1098	64.2	1440	525		0	1.16	1.31	3.46				
June 20 & 21	1600–0800	26–42													223	308	2.0	
June 21	0615	40.3	11.7	155	70	1434	83.8	1680	236	100	0	0.93	1.17	5.04				
	0800–1600	42–50													267	304	10.9	
	1600	50	11.1	44	46	1300	76.0	1538	92	100	0	0.41	1.18	10.47				
June 21 & 22	1600–0800	50–66													263	305	11.6	90D69 100P75S
June 22	0615	64.3	11.2	40	55	1204	70.4	1420	39	100	0	0.46	1.18	8.61				
	0800–1600	66–74													284	304	11.8	145 "
	1600	74	10.8	19	50	1093	64.0	1396	18	152	0	0.36	1.27	10.81				
June 22 & 23	1600–0800	74–90			Feed 72					Feed 75					243	303	6	440 "
June 23	0615	88.3	11.6	32	73	1128	66.0	1299	25	90	0	0.54	1.15	6.71				
	0800–1600	90–98			Feed 67					Feed 91					258	307	5	230 "
	1600	98	11.8	42	90	1203	70.4	1416	45	60	0	0.67	1.17	5.89				
June 23 & 24	1600–0800	98–114			Cond. 6										237	302	5	445 "
June 24	0820	114.3	12.0	75	122	1331	77.8	1690	85	45	0	0.97	1.26	4.86				
	0800–1100	114–117				Secured at 1100 for week-end.									233	308	5	110 "
June 26	1000–1600	117–123													242	306	8	60D44AS
	1600	123	12+	71	136	1297	75.8	1445	102	39	0	1.04	1.12	3.88				
June 26 & 27	1600–0800	123–139													226	304	7.1	210 "
June 27	0620	137.3	11.9	33	127	1340	78.4	1465	24	55	0	0.86	1.09	4.75				
	0800–1600	139–147			Cond. 2										279	300	10	80D47AS
	1600	147	12.0	33	86	1330	77.8	1550	14	24	0	0.61	1.17	7.09				
June 27 & 28	1600–0800	147–163													262	303	6.3	200 "
June 28	0615	161.7	11.6	18	57	1156	67.6	1416	4	20	0	0.40	1.22	9.88	...	...	..	....
	1500	170	12.0	59	96	1461	85.4	1541	67	33	0	0.76	1.05	5.66				
	0800–1515	163–170.2				Secured at 1515.									288	301	3.0	200 "

Determinations in parts per million unless otherwise noted.
Total water evaporated—45,809 lb.
Average evaporation—lb. per hour—269.1.
Average evaporation, lb. per hour per sq. ft. heating surface—16.15.
Hours' evaporation—170.2.

operation. The summarized record of one of the final test runs, as a sample of the completeness of the data collected, is shown in Table I.

Results of the Comparative Tests

The methods of feedwater treatment which were tested may be divided into four general types. These are the colloidal, the coating, the electrolytic and the chemical methods. The colloidal method consists of the use of a colloidal, or semi-colloidal, dispersion of material, to which some inorganic treating-chemicals usually are added. Treatment by coating is designed to provide such a protected evaporative surface that the scale either will be prevented from adhering, or will be removed easily by cleaning. The electrolytic method consists of the maintenance of a small, direct current from a replaceable anode in the boiler drum to the boiler shell, which is connected as the cathode. The chemical method employs mixtures or combinations of recognized, inorganic, treating chemicals, such as soda ash or trisodium phosphate, either alone or together with small quantities of tanning or other organic materials. The results obtained will be discussed by considering each type of treatment separately.

Five colloidal methods of treatment were tested. The colloidal matter for three of them was derived from sea weed, the fourth was extracted from a seed and the fifth consisted of an oxidizable, heavy metal, which was dispersed to colloidal dimensions and suspended in water. In general, the use of these materials produced most desirable sludge characteristics and floc-formation. In the cases where the acid character of the feedwater was recognized, and proper provision was made for the continuous maintenance of alkalinity, the corrosion results were satisfactory. The boiler water in all cases was examined under the ultra-microscope and, although evidence of colloidal motion was discovered, in some cases it was very slight.

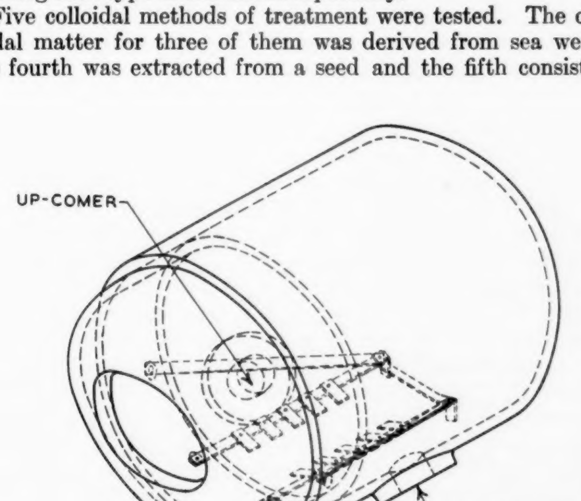

Fig. 3—Arrangement of corrosion specimens within the boiler drum

The principal difficulty of the colloidal methods was in the matter of control. Colloidal methods in general did not provide a direct, analytical determination of the colloidal material in the boiler water. The control therefore had to be based on secondary determinations. One of the most useful of these secondary determinations is that of the sulphate-chloride ratio. Rapid and sufficiently accurate methods are available for the determination of both sulphate and chloride. Under uniform feedwater conditions the maintenance of the same sulphate-chloride ratio in the boiler water as obtains in the feedwater insures that no scale-forming sulphates have precipitated, but that all have been retained in solution.

The exact action of colloidal matter in boiler-water treatment still is open to conjecture. Perhaps one of the most reasonable explanations is that the colloidal action is of the reactive type; in other words, the colloidal material acts as would a properly applied inorganic chemical. This theory was found to apply quite conclusively in the case of an emulsion type of treatment, whose action supposedly is purely colloidal. The precipitate in this test analyzed high in carbonate with some phosphate. The use of the sulphate-chloride ratio is further evidence that the scale-preventing action of the colloids is similar to that of ordinary treating chemicals.

The work done with the materials which for their success depend upon maintaining on the metal a film which resists scale deposition, demonstrated that the theory of scale deposition advanced by Hall,[5] and verified and amplified by Partridge[6] is sound and correct. From this theory it follows that there is no mixture or compound which merely, by its physical presence on the evaporative surface, will prevent scale deposition. Similarly, in the case of tubes made of corrosion-resistant

[5] Hall, R. E., and others, "A Physico-Chemical Study of Scale Formation and Boiler-Water Conditioning," Carnegie Inst. Tech., Min. and Met. Inv., Bulletin 24 (1927).

[6] Partridge, Everett P., University of Michigan, Dept. of Eng. Res., Bulletin 15 (1930).

metals, scale deposition takes place with the same ease as on an ordinary tube.

The test of the electrolytic method was included only to make the investigation more complete. It is doubtful whether the introduction of an additional electrical circuit, particularly one which involves a ground on the ship, ever would be approved for a fighting ship.

The chemical methods which were tested followed the conventional lines. The conversion of the scale-forming salts was accomplished by reaction with carbonate or phosphate. One method made insufficient provision for the counteraction of the acidity of the feedwater, and the boiler water became acid. Although at all times there was present a large excess of soluble phosphate, the sulphate-chloride ratio of the boiler water declined steadily, while the hardness increased. In this case, when the alkalinity was corrected, the boiler-water conditions normally to be expected were restored. At the end of the run a heavy, hard scale, which consisted largely of calcium phosphate, was found in the boiler. This again emphasized the necessity for the maintenance of alkalinity with any treatment in order to obtain satisfactory results.

The nature of the corrosion of boiler metal was studied during the course of the investigation. All of the specimens exposed in the boiler water during the various tests were cut from the same sheet. Four different types of surface were used, namely, as received with mill scale intact; sand blasted, to remove all mill scale; ground to a smooth even surface; ground and polished to give a high surface finish. These specimens were weighed and measured carefully before and after exposure. The loss in weight was calculated to a rate of penetration in inches per year. Only the specimens which had ground surfaces gave results of sufficient consistency to be comparable. The effect of the surface on the consistency of the rate of corrosion is illustrated by the comparative data from two runs, which are shown in Table II.

The corrosion conditions in the boiler were as severe as possible. Not only was a large portion of the chloride in the contaminating solution supplied by magnesium chloride from the Severn River water, but the cold feedwater was saturated with air. Under these conditions the methods of treatment, which were considered to be satisfactory from the point of view of corrosion prevention gave rates of penetration of less than 0.004-in. per year.

The specimens which were on the side of the rack toward the incoming feedwater had a rate of penetration about 75 per cent greater than that on the side above the upcomer. This effect is illustrated by the results shown in Table III.

The results from the corrosion specimens supplied an important, additional reason for emphasizing the maintenance of proper alkalinities in boiler-water treatment. In all satisfactory tests it was found that an alkalinity range from about 0.4 to 0.9 per cent normal was used. It is possible that a wider range will give equally as good corrosion prevention, but it is considered that these limits are also about the best for effecting the conversions required for scale prevention.

Development of Navy Boiler Compound 1933

The results of the various comparative tests, combined with the picture of conditions in the Fleet which had been gained by communications and by analysis of samples from various vessels, provided a satisfactory background for the planning of the new treatment to be prescribed for use in the Navy.

It is considered that one of the outstanding lessons learned in the investigation was the importance of a proper alkalinity in the boiler water. The desired chemical reactions which occur in boiler-water treatment are of the type which require an alkaline medium. There must be a limit below which the reactions proceed at a relatively slow rate. In a highly contaminated water, therefore, scale deposition might occur with all factors in the boiler-water treatment correct except for a slight deficiency in alkalinity. For this reason considerable study was made of all the results obtained in order to decide the limits to be prescribed for the Naval Service. At the same time much consideration was given to the type of test which should be used. The use of pH instead of a titration for alkalinity was considered, but was not adopted because it was not sufficiently sensitive in the desired range of only 11.2 to 12.4. While the use of a titration for alkalinity was continued, phenolphthalein was substituted for methyl-orange indicator in order to increase the accuracy of the results. Alkalinity limits of 0.4 per cent to 0.7 per cent normal (11.7 to 20.4 gr. per gallon) were prescribed.

TABLE II—INFLUENCE OF SURFACE OF SPECIMEN ON CONSISTENCY OF CORROSION RESULTS

	Mill-Scale		Sandblasted		Ground	
	B-9	C-7	B-9	C-7	B-9	C-7
Penetration in inches per year based upon hours' evaporation	0.00150	0.00727	0.000997	0.00246	0.00137	0.00408
	0.00249	0.00452	0.000735	0.00012	0.00180	0.00329
	0.00559	0.00316	0.001798	0.00109	0.00175	0.00392
	0.00134	0.00128	......	......	......	......
Average penetration	0.00273	0.00406	0.001177	0.00122	0.00164	0.00376
Maximum deviation from average	0.00286	0.00321	0.000621	0.00124	0.00027	0.00047
Minimum deviation from average	0.00024	0.00046	0.000180	0.00013	0.00011	0.00016
Mean deviation from average	50%	45%	35%	68%	11%	8%

It was believed that a single compound, which contains the various necessary materials in the proper proportion, is best suited to the needs of the Navy. This opinion was based, not only upon a recognition of the general uniformity of feedwater conditions in the Service, but also upon an appreciation of certain psychological factors in the administration of water treatment.

During the early progress of the investigation the original Navy Standard Compound was modified for the use of the Fleet with the idea of getting results from service tests prior to the adoption of the final formula, and also to provide additional information from selected ships as to the conditions to be met. The modified compound in some cases proved satisfactory for existing conditions, while in other cases it was somewhat deficient. This was a fortunate circumstance as it provided information vitally necessary for completing the development test work. The original contaminating solution was altered to give a greater proportion of sulphate to chloride and the rapidity of feeding was increased about thirty per cent. This also permitted reducing the length of the final tests to 120 hours.

TABLE III—INFLUENCE OF LOCATION ON RATE OF CORROSION OF SPECIMENS

Run	Locations 1–5	Locations 6–10
C-14	0.00166	0.00216
C-15	0.00235	0.00425
C-17	0.00611	0.01379
B-18	0.00225	0.00231
B-20	0.0108	0.0182
Average	0.00462	0.00814

Most boiler-water treatments do not provide means for minimizing carry-over or controlling foaming and priming. Chemical methods usually do not concern themselves with producing a non-adherent sludge. Experiments were conducted with the purpose of incorporating, in the treatment selected, materials which would provide for both of these conditions. Corn starch was tried as an inexpensive, stable, organic material. The results obtained were most gratifying as it was found that starch had qualities for boiler-water treatment far beyond expectations. First, starch is a powerful deterrent to foaming, priming and carry-over. In a test with one of the Station boilers a carry-over which produced 5.6 gr. per gallon of chloride in the condensate was reduced, about 94 per cent, solely by the addition of starch. Second, the starch makes the boiler sludge non-adherent. If the sludge is allowed to dry after the boiler is emptied, it cracks into small blocks which can be brushed out easily, or, if the deposit is thin, can be

blown out with air. Third, the action of the starch improves the coagulation to a marked degree. Fourth, when scale is present in a boiler and is in process of removal by the treatment in use, the action apparently is accelerated. It is thought that this is caused by the effect of the starch on coagulation. As soon as conversion takes place on the scale surface, the precipitate is lifted off and a fresh scale surface is presented for attack by the treating chemical. Fifth, in cases where the treatment is insufficient or ineffective, the presence of starch in the scale deposits makes them softer and more easily removable than is the case in the absence of starch. It appears that the use of starch should decrease trouble from fouling of turbines, both by the reduction of carry-over, and by decreasing the hardness and adherence of such deposits as cannot be prevented.

It was now possible, by combining the available information and experience gained from tests of commercial methods of treatment, the data obtained from service tests of the modified compound, and the results from the investigation of the behavior of starch, to formulate a mixture suitable for use on all naval vessels. Mixtures of trisodium phosphate or disodium phosphate, soda ash and corn starch were used under the conditions of the accelerated standard test. It had been noted previously that the starch seemed to deter the hydrolysis of the carbonate. This was found to be the case in the final runs made at working pressures of 175, 325, 450 and 600 lb. This stabilization of the carbonate allows a considerable portion of the soda ash to be used as the scale-preventing chemical, and also permits the use of an alkalinity test as a method of control, both for corrosion limits and as a measure of the scale-preventing chemicals present.

The formula recommended for use consists of anhydrous disodium phosphate, soda ash and corn starch in the proportions of 47 per cent, 44 per cent and 9 per cent, respectively. This form of phosphate is considered the most economical one, both as to initial cost and as to the space and weight required for chemically equivalent quantities.

No method of treatment will give good results unless it is used properly. A new testing cabinet, which includes the apparatus necessary for determination of alkalinity, chloride and hardness, was designed for use with the improved treatment. With this equipment the necessary analyses may be performed more easily, more rapidly and more accurately than had been possible previously. Complete instructions regarding the purpose and control of the new treatment have been detailed in a revised edition of Chapter 6 of the "Manual of Engineering Instructions."

At this time the new treatment has been used on only one ship, which was known to have somewhat worse feedwater conditions than those usually encountered. The results to date have been satisfactory. It probably will be six months or a year before more definite information will be available.

It may be that in spite of the care in its testing, Navy Boiler Compound 1933 will not provide sufficient protection against scale formation. Conversely, as the existent scale is removed, the Compound may introduce an unwarranted and wasteful excess of materials to the boilers. In either case it will be a simple matter to alter the relative proportions of the three ingredients, basing the change on the results of this investigation. Certainly the boiler operators will be secure in the knowledge that their boilers are cleaner.

Correction

In the article on "Plate-Type Air Heater Development and Present Status in the United States" by W. S. Patterson, in the November issue of COMBUSTION, there was an error in the placing and title of Fig. 5. The gas side should have been on the right and the air side on the left. Moreover, the cut on the right should have been reversed so as to show the flow upward.

Combustion Engineering Announces Formation of Marine Department

Combustion Engineering Company, Inc., announces the formation of a Marine Department and the appointment of Commander Horace T. Dyer as manager. This department will take over the work previously handled by the Marine Division of Combustion Engineering's subsidiary the Hedges-Walsh-Weidner Company and will provide a complete service to the marine field in connection with the company's sectional header, bent-tube and other types of boilers and related equipment.

Commander Dyer is a graduate of the United States Naval Academy, Class of 1907, and was for many years in the engineering branch of Navy work until his resignation in 1923. Subsequently he was Chief Engineer of the Peabody Engineering Company and for the past three years was associated with Gibbs & Cox, naval architects and marine engineers.

Mr. James S. Malseed, marine representative of the Hedges-Walsh-Weidner Company at Washington, will continue as marine representative of Combustion Engineering Company, Inc., with offices in the Mills Building, 17th Street at Pennsylvania Avenue.

Combustion Engineering has made many outstanding boiler installations and its plants are equipped to build the most modern types of high-pressure, high-capacity boilers of either riveted-drum or fusion-welded construction. It has specialized in boiler development along these lines and is prepared to cooperate with naval engineers and architects in the adaptation of higher pressures and temperatures to marine applications.

An Offering to Engineering Colleges

The Meriam Company of Cleveland, Ohio, which manufactures manometers and flowmeters for measuring the pressure and flow of liquids and gases in pipes, has announced that it will present free upon request, to engineering colleges and educational centers having engineering courses, one of its latest type "clean-out" manometers. This instrument, if used in conjunction with a suitable orifice disk in the pipe line, may be used for calculating the velocity or flow in cubic feet or gallons per minute. The relative specific gravities of any two nonmiscible liquids may also be determined. Copies of the manual, "The Manometer and Its Uses," will also be sent to those interested.

A similar offer of standard U-tube manometers was made in 1930 which resulted in 163 such instruments being donated to colleges.

Repairing Drums Weakened by Embrittlement

The expedient of applying wider butt straps as a repair of a seam that had been weakened by caustic embrittlement, was made use of recently on batteries of water-tube boilers at two different locations in southern states. This form of repair is limited, of course, to cases where the embrittlement is in an early stage, and where the design of the boiler does not prevent the use of wider straps; but in the cases described it was successfully used in adding several years of life to boilers that otherwise seemed destined for a quick retirement. All the boilers were of the bent-tube type, and embrittlement had affected only the longitudinal seams in their lower drums.—*The Locomotive.*

Burning Characteristics of Pulverized Coals and the Radiation from their Flames*

Laboratory investigations at Battelle Memorial Institute on Hocking, Pocahontas, Illinois No. 6 and Pittsburgh No. 8 coals show, among other things, that fineness of grinding becomes increasingly important as the combustion space is restricted; that the type of coal influences the rate of combustion; that increased furnace temperature increases the rate of combustion; and that, although the temperature and total radiation of the flame are affected by fineness of grinding, excess air and rate of firing, the emissivity of the flame at any position is affected to a marked degree only by the type of coal.

By

RALPH A. SHERMAN

Fuel Engineer
Battelle Memorial Institute
Columbus, Ohio

FOR some time, the Battelle Memorial Institute has conducted an experimental investigation of the combustion of pulverized fuels in a large-scale laboratory apparatus. The object of this investigation was to determine the relation of the rate of burning of pulverized fuels and of the radiation from their flames to the type of fuel and the fineness of pulverization.

A previous paper[1] discussed the work of other investigators, described in detail the apparatus and the methods of testing, and presented data on the progress of combustion and the amount of carbon remaining unburned at various points in the flame when burning Ohio No. 6 or Hocking and Pocahontas No. 3 coals and some semi-cokes made from Hocking coal. The conclusions presented in that paper were:

1. The primary product that appears in the combustion of fuel in pulverized form is CO_2. With ratios of air to coal equal to or greater than required, CO appeared only in small amounts in the initial carbonization of the coal. CO found in industrial furnaces burning pulverized fuel is undoubtedly formed in the stream of coal and primary air which is generally much less than the amount required for complete combustion; the CO once formed may persist for a considerable distance of flame travel because of slow mixing with secondary air.

2. With Hocking coal at 8.5 or 10.5 feet from the burner the unburned carbon loss did not decrease with increase of excess air above 20 per cent.

3. No one numerical value that fully expresses the fineness of pulverized fuel with significance in regard to its combustion characteristics was found. The specific surface did not appear to have greater general significance than the percentage passing 200-mesh. For equal percentages passing 200-mesh the amount retained on 100-mesh was markedly significant. Pulverized fuel equipment manufacturers would undoubtedly do well to develop equipment that would deliver a uniformly-sized product rather than one of extreme fineness.

4. The rates of combustion and percentages of unburned carbon varied greatly with the type of fuel. The loss in unburned carbon was greater for Pocahontas than Hocking coal at equal percentages through 200-mesh and other conditions equal, although the "superfines" and specific surface were much greater for the Pocahontas coal. A wide enough range of fuels has not yet been covered to develop a relation between the burning characteristics and other known properties of the fuels.

5. Semi-cokes having contents of volatile matter as low as 12.8 per cent burned satisfactorily as pulverized fuel, although their carbon loss was higher for similar conditions than that of the coal from which they were made.

6. Pulverized fuels were burned in the test furnace at rates of heat liberation equal to those of industrial furnaces. The unburned carbon losses after only 10.5 ft. of flame travel and with a burning time of approximately 0.3 sec. were equal to those attained with longer flame travel in industrial furnaces. This was attained with a simple burner which admitted all the air with the coal and with no attempt at turbulence. If this principle were extended to burners of high capacity, it would not work as well, because conditions would not be as favorable for ignition of the fuel, because the volume of the cone of fuel would increase so much more rapidly than the surface. Division of the coal and air among a large number of small burners would offer certain mechanical difficulties, as (1) the uniform distribution of coal and air among a number of burners, particularly in a direct-fired unit, (2) the increased power consumption for blowing air through small ducts with the coal, and (3) the increased difficulties in the use of preheated air.

Scope of Paper

The present paper includes further work that has been done on the Hocking and Pocahontas coals and on Illinois No. 6 and Pittsburgh No. 8 coals. Data are presented:

1. On the relation of the excess air to the combustion of Hocking and Pocahontas coals.
2. On the relation of the fineness of grinding to the combustion of the four coals.
3. On the relation of the rate of firing and the furnace temperature to the completeness of combustion.
4. On the radiation and emissivity of the flames and their relation to the excess air, fineness and rate of firing for the four coals.

* Prepared for presentation, under auspices of Fuels Division, at the Annual Meeting of the American Society of Mechanical Engineers, December 4–8, 1933.

[1] Sherman, R. A. Proc. Third Int. Conf. on Bituminous Coal, 1931, Vol. II, p. 510.

Apparatus and Methods

The testing apparatus included a drier, conical ball mill, air classifier, storage and weighing bins, screw feeder, fan, burner, furnace and equipment for sampling gases and solids and for the measurement of temperature of the gases and their radiation. As the apparatus and methods, except for the radiation work, were described in detail in the previous paper, they will not be repeated.

TABLE I.—CHARACTERISTICS OF COALS

Coal	Ohio No. 6 Hocking	Pocahontas No. 3	Illinois No. 6	Pittsburgh No. 8
Source				
State	Ohio	West Virginia	Illinois	Ohio
County	Athens	Mercer	Jackson	Belmont
Mine	Lick Run	Louisville	Kathleen	Blaine
Composition, as fired				
Proximate				
Moisture	2.3	0.4	1.1	0.5
Volatile matter	35.4	17.2	35.6	40.0
Fixed Carbon	52.3	76.6	50.4	50.2
Ash	10.0	5.8	14.9	9.3
	100.0	100.0	100.0	100.0
Ultimate				
Carbon	71.8	85.0	66.7	72.3
Hydrogen	5.1	4.4	4.6	5.0
Oxygen	10.3	2.2	11.0	7.4
Nitrogen	1.3	2.0	1.5	1.2
Sulphur	1.5	0.6	1.3	4.8
Ash	10.0	5.8	14.9	9.3
	100.0	100.0	100.0	100.0
Calorific value, B.t.u. per lb.	12,420	14,790	11,770	13,200
Softening temp. of ash, fahrenheit	2,590	2,630	2,250	1,985
True specific gravity	1.42	1.33	1.46	1.40

For convenient reference, however, Fig. 1 shows details of the furnace and burner. The furnace was tightly sealed and all the air used for combustion was supplied through the burner with the coal. The purpose was to study the combustion in a stream of coal and air of uniform mixture and samples were taken on the center line of the furnace as the closest approximation to the desired ideal. The apparatus and methods for the determination of the radiation from the flame are described later in this paper in the section that deals with that work.

Characteristics of Coals

Table I gives the source, proximate and ultimate analyses, calorific value, softening temperature of ash and true specific gravity of the coals burned; the analyses are on the basis of typical moisture as fired.

Table II gives typical size characteristics of the pulverized coals as determined by screen tests and by microscopic counting. The pulverizing equipment, ball mill and classifier, gave a lower percentage of coarse particles, +30 or 50-mesh sieve, than is frequently obtained in industrial practice. Thus, with 75 per cent or more passing the 200-mesh sieve, not over 0.2 per cent remained on the 50-mesh sieve. The importance

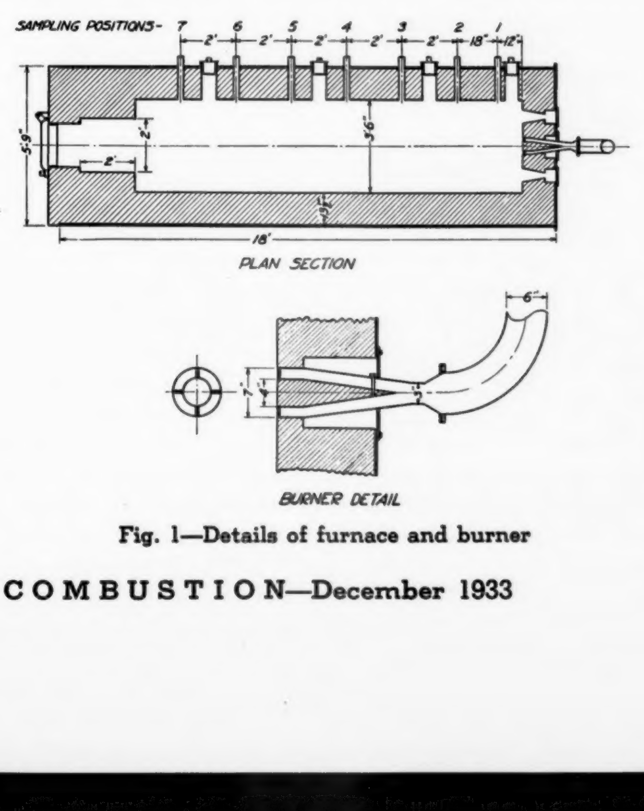

Fig. 1—Details of furnace and burner

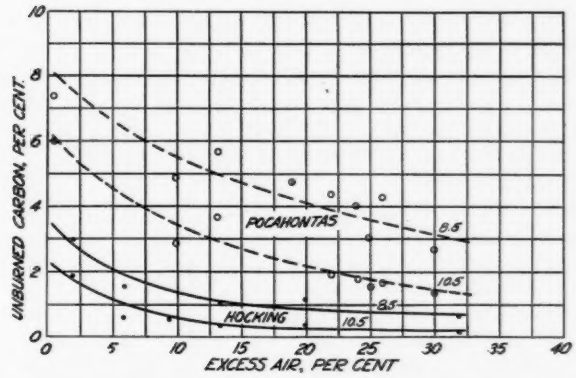

Fig. 2—Relation of unburned carbon to excess air, Hocking and Pocahontas coals. Through 200 mesh, 80 per cent; rate of heat input 2,650,000–2,800,000 B.t.u. per hour. Figures on curves are distances from burner in feet

of the elimination of the coarse sizes was shown by data in the earlier paper.

TABLE II.—SIZE CHARACTERISTICS OF COAL

	Cumulative Oversize, Per Cent				−200, Per Cent	Mean Diam., Mm.	Total Surface, Cm.²/gm.	Surface in −200 Mesh Coal Per Cent
U.S.S. Sieve No.	30	50	100	200				
Size Opening, mm.	0.590	0.297	0.149	0.074				
Hocking								
	0.2	2.0	19.8	56.6	43.4	0.053	790	79.1
	0.0	0.2	7.8	44.2	55.8	0.035	1215	88.2
	0.0	0.0	0.6	24.6	75.4	0.032	1325	93.4
	0.0	0.0	0.4	19.8	80.2	0.028	1514	95.4
Pocahontas								
	0.6	9.0	31.4	61.6	38.4	0.039	1155	85.6
	0.0	0.2	9.2	39.2	60.8	0.025	1800	92.6
	0.0	0.0	1.0	21.0	79.0	0.012	3615	97.7
	0.0	0.0	0.6	9.0	91.0	0.010	4685	99.3
Illinois								
	0.2	2.0	23.0	60.6	39.4	0.059	695	75.6
	0.0	0.6	10.2	48.8	51.2	0.044	930	83.6
	0.0	0.2	7.4	44.2	55.8	0.041	1005	85.9
	0.0	0.2	1.2	24.0	76.0	0.033	1245	92.1
	0.0	0.0	0.4	15.0	85.0	0.027	1535	96.7
Pittsburgh No. 8								
	0.2	1.8	19.0	57.8	42.2	0.055	765	77.3
	0.0	0.8	10.6	48.2	51.8	0.048	900	82.9
	0.0	0.2	4.6	35.0	65.0	0.037	1175	90.0
	0.0	0.0	0.2	14.8	85.2	0.022	1925	97.2

The size characteristics of the three high-volatile coals were similar but the Pocahontas coal was marked, not only by the greater fineness shown by the smaller average diameter or greater specific surface, but also by the larger percentage of the coarse sizes.

Discussion of Combustion Data

Excess Air and Unburned Carbon. Fig. 2 shows the relation of the percentage of carbon remaining unburned at 8.5 and 10.5 ft. from the burners to the percentage of excess air when burning Hocking and Pocahontas coals; the data for the Hocking coal are those presented in Fig. 17 of the previous report. The unburned carbon continued to decrease up to 30 per cent excess air with the Pocahontas coal, whereas the curves for the Hocking coal were practically flat beyond 20 per cent.

Fineness of Grinding and Unburned Carbon. Figs. 3 to 6 show the relation of the percentage of carbon unburned at various points in the flame to the percentage of the coal passing the 200-mesh sieve for the four coals when burned at similar rates of heat input and with 20 per cent excess air. The percentage through 200-mesh is used as a basis for plotting for, as stated above in conclusion 3 from the previous paper, no better basis had been found. The data for the Hocking coal are based on coal of the fineness characteristics given in Table II; the unburned carbon for coal of 70 per cent or more

through 200-mesh sieve is, therefore, less than that shown in Fig. 15 of the first paper.

The curves show the great importance of fineness of pulverization as the combustion space is restricted; an exception was the Pocahontas coal at 4.5 ft. from the burner where increased fineness apparently did not decrease the amount of unburned carbon.

Although they do not break sharply, the curves for 10.5 ft. from the burner tend to become flat and indicate approximately the limit beyond which increased fineness results in little decrease in unburned carbon. These limits are approximately 70 per cent for Hocking, 85 per cent for Pocahontas, 75 per cent for Illinois and 65 per cent for Pittsburgh coal. Obviously, for a particular installation the optimum fineness of pulverization must be determined by an economic balance between the decrease in the loss in unburned carbon and the increased power for finer pulverization. No power consumption data were taken in this investigation.

Relation of Type of Coal to Unburned Carbon. Figs. 3 to 6 also indicate the relative rates of combustion of the four coals but Fig. 7, in which the unburned carbon and temperature are plotted against the burning time, shows the relation more clearly. The times were calculated from the velocity readings taken with a water-cooled pitot tube on the center line of the furnace. Although only approximate, they afford better comparisons than distance only.

The curves for the Hocking and Pittsburgh coals are practically coincident; they burned much more rapidly than the other two coals and Pocahontas coal burned much the slowest of the four coals. The temperature curves show similarly the difference in the rates of burning. The temperature at the first position was lowest for the Pocahontas coal and it increased most slowly. The greatest difference in the final temperature was about 100 fahr.; it was highest for the Pocahontas coal and lowest for the Illinois coal.

That Pocahontas and other coals of similarly low content of volatile matter burn more slowly in pulverized form than coals of higher volatile content is well known. An obvious explanation is the fact that, because less carbon is distilled in gaseous hydrocarbons, more must be burned as solid carbon which burns less rapidly than the gas.

However, these data show that the rate of burning is not directly proportional to the content of volatile matter. The Illinois coal burned more slowly than the Hocking coal, although their volatile contents were practically the same on the as-fired basis and that of the Illinois coal was 2 per cent greater on the moisture- and ash-free basis. Pittsburgh coal burned only slightly more rapidly than Hocking coal, although their volatile contents on the moisture- and ash-free basis were 44 and 40 per cent, respectively.

The ash content, oxygen content, quality of volatile matter

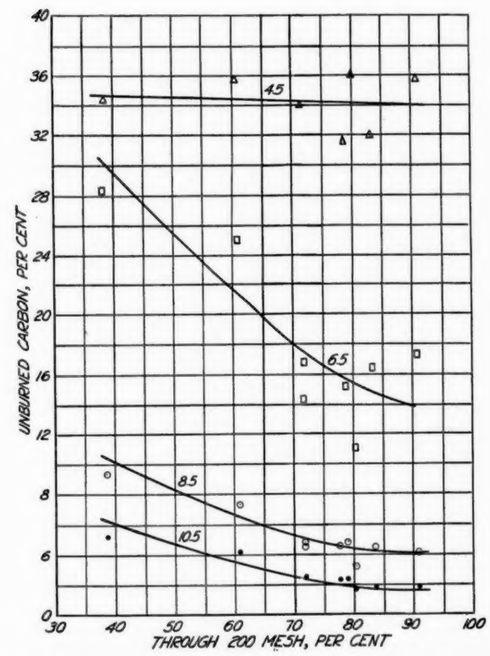

Fig. 4—Relation of unburned carbon to fineness of Pocahontas coal. Excess air 20 per cent; rate of heat input 2,780,000 B.t.u. per hour. Figures on curves are distances from burner in feet

and probably other factors undoubtedly control the rate of burning as well as volatile matter content and many coals would have to be investigated before even an approximate relation were found between the rates of burning and the composition of the coals. Godbert[2] found that a reactivity index was closely related to the burning time. It is also considered possible that the ignition temperature of the coal may be an important factor if it were determined under conditions of actual suspension of the coal in an air stream. The initial rate of combustion is so rapid that a small gain in time by earlier ignition would make a great difference in the rate of combustion as measured from the time of entrance of the coal into the furnace.

Relation of Rate of Combustion to Furnace Temperature. Griffin, Adams and Smith[3] in a series of small-scale experiments on the burning of individual particles of fuel found, among other results, that the burning time increased with increase of temperature.

An explanation of this surprising result has been proposed by Burke and Schumann[4] as due to the fact that the mass of oxygen per unit volume of the film surrounding the particle decreases with increase of temperature.

The possible change in the temperature of the large-scale furnace was not large. It could be obtained in two ways: (1) by taking data at varying intervals from initial lighting of the furnace and (2) by change in the rate of firing.

Figs. 8, 9 and 10 show, respectively, for Hocking, Pocahontas and Illinois coals, the relation of the unburned carbon to the burning time with varying furnace temperature. The different temperatures were obtained by taking observations at 2, 5 and 7 hours after lighting the furnace. The increase in the maximum temperature with increase in time from lighting the furnace was about 200 fahr. with each coal. As the temperature of the gases increased, their velocity increased and the time in the furnace decreased.

The actual difference in unburned carbon with difference in time was greatest for the Pocahontas coal. The difference decreased with increasing distance from the burner. The general

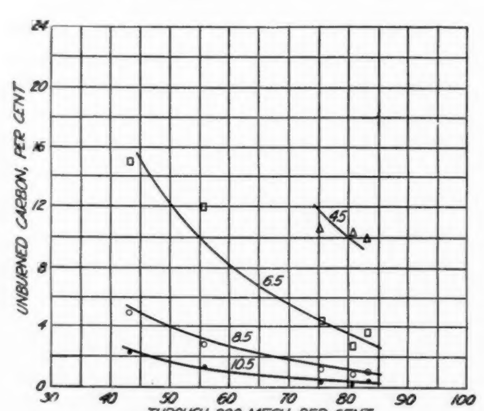

Fig. 3—Relation of unburned carbon to fineness of Hocking coal. Excess air 20 per cent, rate of heat input 2,650,000 B.t.u. per hour. Figures on curves are distances from burner in feet

[2] Godbert, A. L. Fuel, 9, 1930, p. 57.
[3] Griffin, H. K., Adams, J. R., and Smith, D. F. Ind. Eng. Chem., 21, 1929, p. 808.
[4] Burke, S. P., and Schumann, T. E. W. Ind. Eng. Chem., 23, 1931, p. 406.

similarity of the slope of the three curves for each coal again suggests that the principal difference may have been in the time of ignition of the coal. As the furnace became hotter near the burner, the ignition occurred earlier and this again increased the temperature in the front part of the furnace.

Fig. 11 shows the effect of temperature on the rate of burning of Illinois coal when the change in temperature was effected by change in the rate of heat input. The amount of unburned carbon at each position increased with increase in the rate of firing, but, as the velocity also increased, these curves show that the burning time to a given percentage of unburned carbon decreased with increase in the rate of firing.

These results show that in this furnace, and by analogy, undoubtedly, in large boiler furnaces, the burning time decreased with increase in furnace temperature rather than increased as indicated by the small-scale experiments to which reference was made. It will be seen, however, that the greatest difference was, in general, in the early part of the flame and the slope of the curves for the latter part of the travel was so similar that the principal difference may have been the time of ignition of the coal. As the walls of the furnace near the burner became hotter with increase in time or increase in rate of firing, the ignition occurred earlier.

A similar change in the temperature of a boiler or other furnace would have a similar effect on the rate of ignition; therefore, the desirability of maintenance of a hot zone near the burner is obvious. However, it cannot be concluded that water-cooled boiler-furnace walls will, in general, decrease markedly the overall rate of combustion. In large furnaces the flames are so thick that, as shown later in this paper, they become practically black and the cold wall does not so greatly affect the temperature at the burner as it did in the experimental furnace. In small furnaces, such as those of Scotch Marine or domestic boilers, the temperature of the walls has an important effect on the temperature in the flame and, therefore, on the combustion process.

Radiation from Pulverized-Coal Flames

The percentage of the heat liberated in a furnace that is transferred to the heat-receiving surface by radiation is an important factor in the design of boilers, oil stills and similar

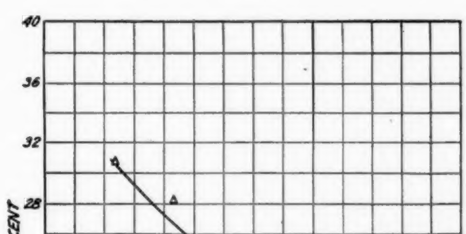

Fig. 5—Relation of unburned carbon to fineness of Illinois coal. Excess air 20 per cent; rate of heat input 2,650,000 B.t.u. per hour. Figures on curves are distances from burner in feet

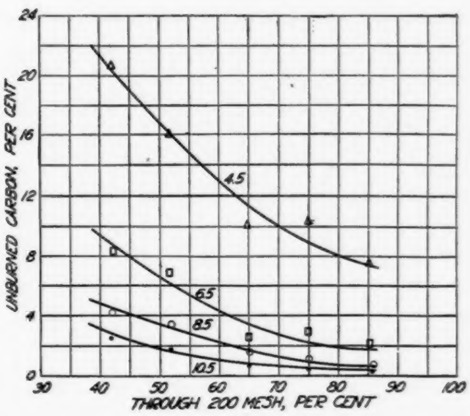

Fig. 6—Relation of unburned carbon to fineness of Pittsburgh No. 8 coal. Excess air 20 per cent; rate of heat input 2,700,000 B.t.u. per hour. Figures on curves are distances from burner in feet

equipment, as it determines the desirable relation of the radiation and convection surfaces. Broido,[5] Orrok[6] and De Baufre[7] have considered empirical formulas, and Wohlenberg[8] and co-workers, and Haslam and Hottel[9] have proposed formulas of more fundamental basis for the calculation of the fraction of the total heat liberated in the furnace that is radiated to the tubes in a boiler or other furnace. Experimental determinations of the radiation from the flame have been lacking. Koessler[10] has published the results of the determination of the radiation in several types of furnaces.

Apparatus and methods were developed to measure the radiation from the flames in the experimental furnace in the hope that the results could be interpreted for other furnace conditions and that they would serve to determine the validity of the formulas that have been set up.

Apparatus and Method

Fig. 12 shows the arrangement of the apparatus used for the measurement of the radiation from the flame, which was similar to that used by Koessler.[10] It consisted of a Moll thermopile mounted on the end of a water-cooled tube which contained four diaphragms to limit the angle of vision. Around the thermopile was a water jacket through which the water flowed in series with the tube; in this way the cold ends of the thermopile junctions and the surface of the tube that the thermopile "saw" were at the same temperature and the output was zero independent of changes in room or water temperature.

A water-cooled copper cone served as a limiting cold screen to insure that the radiation falling on the surface of the thermopile was from the flame only. The cone shape was adopted to obtain more nearly black conditions by decreasing the possibility of reflection from the surface. This limiting screen was adjustable to obtain variable thicknesses of flame.

The entire assembly was mounted in a plate and refractory block and could be changed from one to another of the four ports provided in the furnace.

The water jacket assembly for the thermopile was hinged so that it could be swung back from the end of the tube. The tube containing the diaphragms could then be removed and a thermocouple run into the furnace for the measurement of gas temperature.

The output of the thermopile was read on a semi-precision

[5] Broido, B. N. Trans. A.S.M.E., 47, 1925, p. 1123.
[6] Orrok, G. A. Trans. A.S.M.E., 47, 1925, p. 1148.
[7] De Baufre, W. L. Trans. A.S.M.E., 53, 1931, p. 253.
[8] Wohlenberg, W. J., and Morrow, D. G. Trans. A.S.M.E., 47, 1925, p. 127. Wohlenberg, W. J., and Lindseth, E. L. Trans. A.S.M.E., 48, 1926, p. 849. Wohlenberg, W. J., and Anthony, R. L. Trans., A.S.M.E., 51, 1929, p. 235.
[9] Haslam, R. T., and Hottel, H. C. Trans. A.S.M.E., 50, 1928, p. 443.
[10] Koessler, P. Archiv für Warmewirtschaft, 11, 1930, p. 229.

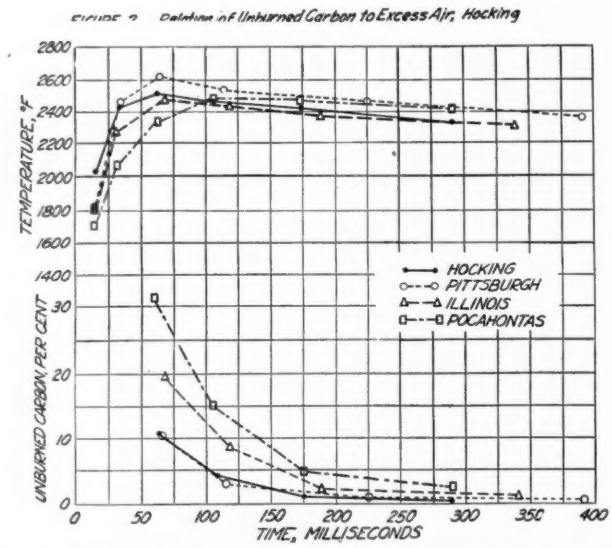

Fig. 7—Comparison of rate of burning of four coals. Rate of heat input 2,650,000–2,710,000 B.t.u. per hour, excess air 20 per cent; through 200 mesh, 75–79 per cent

potentiometer with external reflecting galvanometer; this could be read to one microvolt.

The thermopile was calibrated with water jacket and water-cooled diaphragm tube as a unit. This eliminated the necessity of making any calculations for the correction for the angle of vision. A graphite cylinder heated in a gas furnace was used as a black-body source; a thermocouple on the inside back wall of the body measured its temperature. A straight-line calibration curve was obtained; the factor was 145 B.t.u. per sq. ft. per hr. per microvolt output.

The thermopile attained full output quickly, but had enough lag that it ironed out small fluctuations in the flame radiation. Only with violet fluctuations in the flame temperature did the galvanometer swing so rapidly as to make accurate readings difficult.

Accuracy of Results

The greatest weakness of this work as it is with most similar work was the measurement of the temperature. The exposed thermocouple was subject to gain or loss of heat by radiation, but shielded velocity thermocouples, although tried, were found, as to be expected, impossible to use with pulverized-coal flames because of the accumulation of ash on the wire and in the shielding tube. Because the temperature differences in this furnace were not great and as it was not indicated that any other method would give enough greater accuracy to warrant the difficulties of use, the exposed thermocouple was used.

The radiation from non-luminous flames consists only of the radiation from CO_2 and H_2O. The radiation of these gases has been calculated and determined by Schack[11] and Schmidt;[12] therefore, a comparison of the measured and calculated radiation from non-luminous flames allows some estimation of the order of accuracy of the measurements.

Table III[13] shows a comparison of the calculated and measured radiation from non-luminous natural gas flames in the experimental furnace.

TABLE III.—COMPARISON OF CALCULATED AND MEASURED RADIATION FROM NON-LUMINOUS NATURAL GAS FLAMES

Test	Heat Input Million B.t.u. per Hr.	Excess Air Per Cent	Distance from Burner Ft.	Radiation 1000 B.t.u. per Sq. Ft. per Hr Calc.	Meas'd	Diff.	Per Cent
1	2.71	20	3.5	23.9	23.9	0.0	0.0
			7.5	22.5	23.8	1.3	5.8
			11.5	20.05	21.0	0.95	4.7
2	2.67	14	3.5	22.5	24.7	2.2	9.8
			7.5	21.0	24.1	3.1	14.7
			11.5	19.0	19.4	0.4	2.1
3	2.63	4.5	3.5	22.6	23.9	1.3	5.8
			7.5	25.4	30.7	5.3	20.9
			11.5	27.7	31.0	3.3	11.9
4	2.77	10	3.5	30.2	30.7	0.5	1.8
			7.5	28.3	29.0	0.7	2.5
			11.5	23.1	22.3	0.8	3.7
5	2.17	12	3.5	22.0	25.4	3.4	15.5
			7.5	19.9	23.2	3.3	16.6
			11.5	17.8	17.8	0.0	0.0

By chance, two of the measured values agreed exactly with those calculated and only one measured value was less than the calculated. The other measured values were 2 to 21 per cent higher than those calculated; the findings of other investigators have been similar. The average difference was 7 per cent. For this type of measurement this is considered excellent agreement; if we assume that the calculated values are correct, the probable error in the measurements should be not more than 10 or 15 per cent.

Discussion of Data

The data on the radiation from the flames are presented in a series of curves; in each figure the temperature of the gases, the radiation from the flame, and the emissivity are plotted in the lower, middle and upper parts, respectively, against distance from the burner. The temperatures given are the averages of a number of readings across the flame and are, therefore, mostly lower than those shown in the previous figures which were taken only on the axis of the furnace.

The radiation is that calculated from the thermopile output,

[11] Schack, A. Zeit. für Tech. Physik, 5, 1924, p. 266.
[12] Schmidt, E. Forschungsarbeiten auf der Gebiete des Ingenieurwesen, 3, 1932, p. 57.
[13] Sherman, R. A. Paper given at Meeting of Iron and Steel Division, A.S.M.E., Pittsburgh, Pa., Feb. 17 1933. To be printed in Trans. Iron and Steel Division.

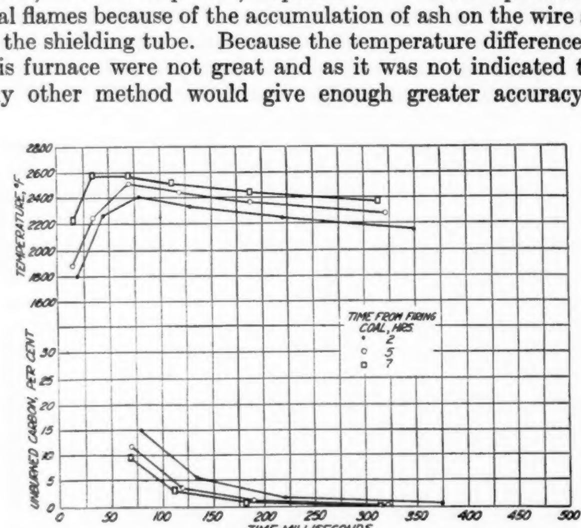

Fig. 8—Relation of rate of burning to furnace temperature, Hocking coal. Excess air 18 per cent; through 200 mesh, 82 per cent; rate of heat input 2,660,000 B.t.u. per hour

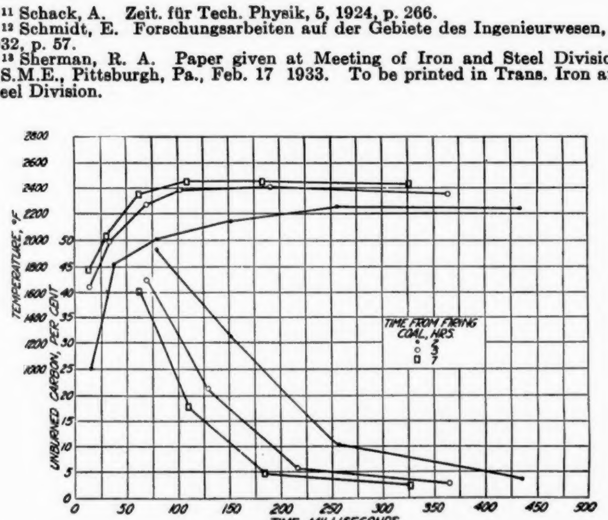

Fig. 9—Relation of rate of burning to furnace temperature, Pocahontas coal. Excess air 20 per cent; through 200 mesh, 78 per cent; rate of heat input 2,800,000 B.t.u. per hour

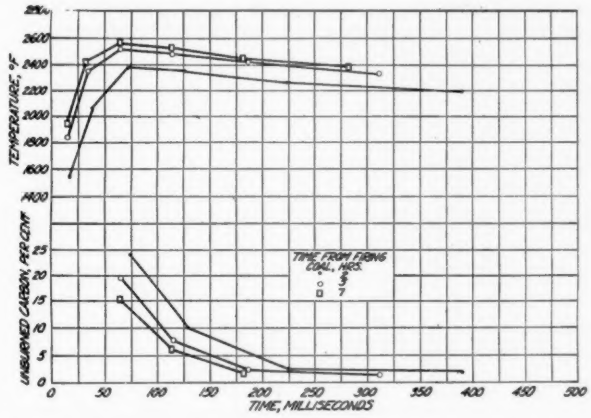

Fig. 10—Relation of rate of burning to furnace temperature, Illinois coal. Excess air, 20 per cent; through 200 mesh, 78 per cent; rate of heat input 2,650,000 B.t.u. per hour

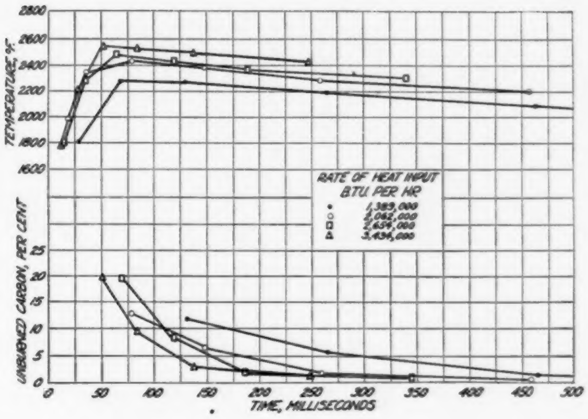

Fig. 11—Relation of rate of burning to rate of heat input, Illinois coal. Excess air, 20 per cent; through 200 mesh, 80 per cent

when 34 or 35 in. of flame were included between the water-cooled back plate and the end of the water-cooled diaphragm tube, and the calibration constant of the thermopile. The emissivity is the ratio of the measured radiation to that of a black body at the temperature of the flame.

Relation of Radiation to Type of Coal. Fig. 13 shows the radiation data for the four coals at similar rates of heat input, fineness and excess air. The curves show that the radiation from the flame does not reach its maximum in the zone of maximum temperature; for example, the maximum temperature with the Pocahontas coal was found at 7.5 ft. from the burner, but the maximum radiation measured was at 3.5 ft. from the burner.

The slow-burning Pocahontas coal had the lowest temperature and radiation of the four coals, except at 11.5 ft. from the burner where both the temperature and radiation were not greatly different for the four coals.

The emissivities of the fast-burning Hocking and Pittsburgh coal flames were lower than those of the other two coals at the first two positions of measurement. The emissivity of the Illinois coal flame was the highest of the four at all positions. However, the maximum difference in emissivities at any position was only 0.10 to 0.15, or about 30 per cent, and the trend for all coals was the same; the emissivity decreased with increasing distance from the burner as the carbon burned out.

Fig. 14 shows a comparison of the temperature, radiation and emissivity of pulverized Hocking coal, and non-luminous and semi- and fully-luminous natural gas flames at similar rates of heat input and excess air. The luminous natural gas flames were obtained by injection of the gas and air in separate streams so that mixing was delayed and the hydrocarbons were cracked to give free carbon. The temperature curves of the pulverized-coal and non-luminous gas flames were similar, but the temperature of the luminous gas flames was low near the burner because of slow combustion.

The radiation and emissivity curves show that, although the non-luminous and semi-luminous natural gas flames had much lower radiation and emissivity than the pulverized-coal flames, yet by a design of burner to give proper cracking natural gas can be made to give flames which have as high radiation and emissivity as pulverized coal.

Radiation and Excess Air. Figs. 15 and 16 show the temperature, radiation and emissivity of Hocking and Pocahontas coal flames with varying excess air. The temperature and radiation both increase as the excess of air decreases, as expected, except that the temperature and radiation were somewhat higher with 10 per cent than with no excess air when burning Pocahontas coal.

No great differences in emissivity with differences in excess air were found and the curves are somewhat intermingled but the emissivities decreased slightly with increase in excess air.

Radiation and Size of Coal. Figs. 17 and 18 show the relation of the radiation to the fineness of grinding for the Illinois and Pittsburgh coals, respectively. Although the temperature and radiation did not increase uniformly with increasing percentages through 200-mesh with either coal, they were greatest with the finest coal. When burning Illinois coal the change in radiation with size of coal was greater than when burning Pittsburgh coal and the emissivity of the Illinois coal flames was 30 to 40 per cent greater with 85 per cent than with 40 per cent through 200-mesh, whereas the emissivity changed only slightly with the size of the Pittsburgh coal.

Radiation and Rate of Heat Input. Figs. 19, 20 and 21 show the relation of the radiation to the rate of heat input for Hocking, Pocahontas and Illinois coals, respectively. The temperatures increased with the rate of firing and the radiation increased in about the same proportion; consequently, the emissivity at any position varied within small limits and not in any definite relation to the rate of firing. As shown in Fig. 13, at one rate of firing, the radiation and emissivity of the Illinois coal flames were higher than those of the other two coals at corresponding rates of heat input.

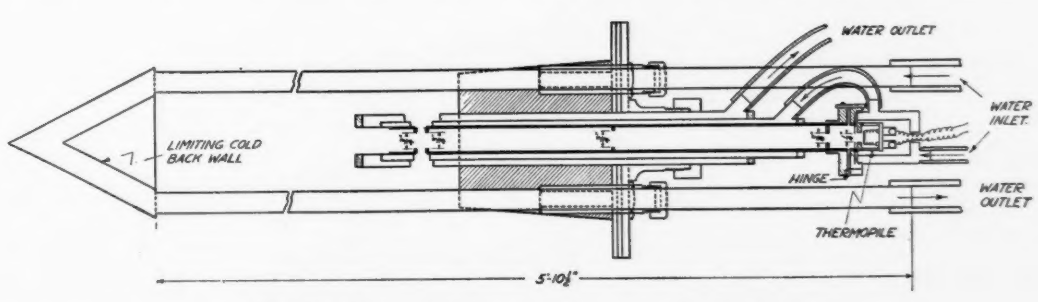

Fig. 12—Apparatus for measurement of radiation

TABLE IV.—COMPARATIVE EMISSIVITIES AND ABSORPTION COEFFICIENTS FOR PULVERIZED-COAL FLAMES

Coal	Hocking	Pocahontas	Illinois	Pittsburgh
Heat Input, B.t.u. per hr.	2,680,000	2,780,000	2,650,000	2,710,000
Excess Air, %	20	20	20	20
Fineness, % −200 Mesh	79.6	73.2	85	85.2

Distance from Burner, Ft.	Depth Flame, In.	p_F	K	Depth Flame, In.	p_F	K	Depth Flame, In.	p_F	K	Depth Flame, In.	p_F	K
0.5	10	0.184	0.0203	10	0.254	0.0293	10	0.172	0.0189	10	0.138	0.0147
	16	.248	.0177	16	.336	.0256	15	.283	.0221	15	.211	.0159
	22	.374	.0212	22	.508	.0323	20	.390	.0247	20	.193	.0106
	28	.586	.0316	28	.680	.0408	30	.680	.0380	30	.586	.0295
	34	.619	.0283	34	.721	.0376	35	.718	.0362	35	.635	.0288
3.5	10	.149	.0161	10	.246	.0283	10	.168	.0182	10	.132	.0143
	16	.246	.0175	16	.366	.0283	15	.260	.0200	15	.199	.0147
	22	.348	.0193	22	.470	.0344	20	.370	.0232	20	.341	.0209
	28	.404	.0187	28	.530	.0266	30	.548	.0265	30	.518	.0244
	34	.458	.0180	34	.568	.0219	35	.601	.0263	35	.504	.0200
7.5	10	.159	.0173	10	.157	.0173	10	.156	.0170	10	.137	.0147
	16	.219	.0147	16	.214	.0150	15	.223	.0168	15	.182	.0136
	22	.270	.0145	22	.253	.0134	20	.279	.0164	20	.220	.0124
	28	.312	.0134	28	.315	.0136	30	.382	.0159	30	.284	.0111
	34	.353	.0129	34	.336	.0120	35	.435	.0164	35	.318	.0108
11.5	10	.147	.0159	10	.246	.0284	10	.153	.0165	10	.088	.0092
	16	.203	.0143	16	.285	.0209	15	.209	.0157	15	.133	.0095
	22	.243	.0129	22	.323	.0177	20	.260	.0150	20	.185	.0104
	28	.290	.0131	28	.359	.0159	30	.353	.0145	30	.259	.0094
	34	.335	.0120	34	.380	.0141	35	.406	.0150	35	.320	.0111

Variation of Radiation with Depth of Flame. As boiler and other furnaces may have flame depths up to 20 ft., it is desirable, if the data obtained in the test furnace are to be of practical value, to be able to calculate the radiation of flames of greater depth. The general expression for the relation of the radiation to the thickness of the flame is

$$Q = Q_B\,(1-e^{-KL}) \qquad (1)$$

where Q is the radiation from the flame, Q_B is the radiation of a black body at the temperature of the flame, e is the base of natural logarithms, K is the absorption coefficient, and L the depth of the flame. As $\frac{Q}{Q_B} = p_F$, the emissivity of the flame, then

$$p_F = 1 - e^{-KL} \qquad (2)$$

As the radiation from these flames is that from CO_2, H_2O and solid carbon and ash particles, the absorption coefficient depends on the partial pressure of the CO_2 and H_2O and on the concentration of the solids in the flame. As neither these nor the temperature of the gas are uniform across the flame at any position, it would not be expected that absorption coefficients, K, calculated from the radiation of various depths of flame, would be constant.

Table IV shows the emissivity p_F and absorption coefficient K calculated for the four coals burned under similar conditions for five flame thicknesses at each of the four points of measurement. The flame thickness was measured in inches for the calculation of K. As expected, K is not constant but the constancy increases with increasing distance from the burner as conditions become more uniform across the flame and, in general, the constancy increases with increase in flame thickness.

To be strictly correct, the emissivity at increasing flame depths cannot be calculated from values of K so determined. Flames containing the amounts of CO_2 and H_2O that these would probably reach their maximum radiation in the wavelength bands of these gases at less depth than the maximum would be reached for the solid particles. Therefore, the solid and gaseous radiation should be separated, but it was not considered that the accuracy of the data would warrant the complications.

Table V gives the calculated emissivities for Illinois coal for increasing flame thicknesses using the K calculated for the maximum depth of measurement from Table IV. For a flame thickness of 10 feet the flame at 0.5 or 3.5 ft. from the burner would be practically black and emissivity almost 1, and at 7.5 and 11.5 ft. the emissivities would be over 0.8. At a flame thickness of 20 ft. the flame would be practically black at all distances from the burner.

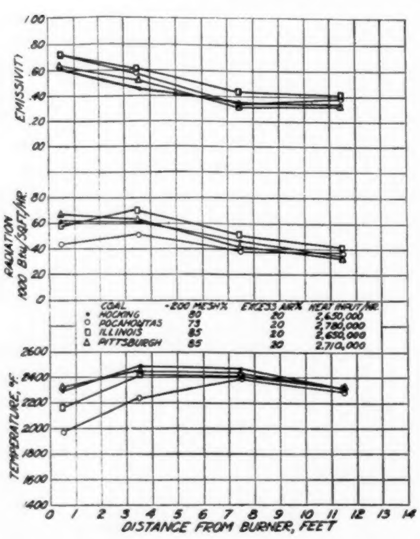

Fig. 13—Comparative temperature, radiation and emissivity of Hocking, Pocahontas, Illinois and Pittsburgh coal flames

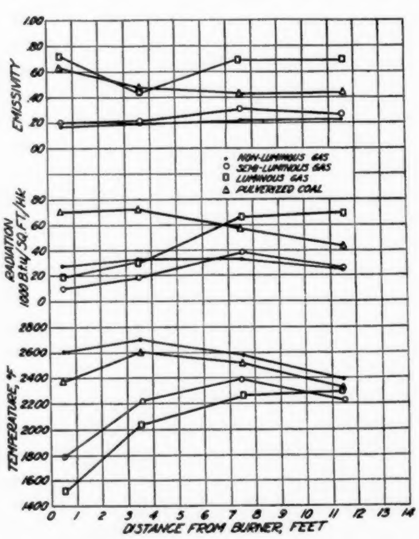

Fig. 14—Temperature, radiation and emissivity of gas and Hocking coal flames

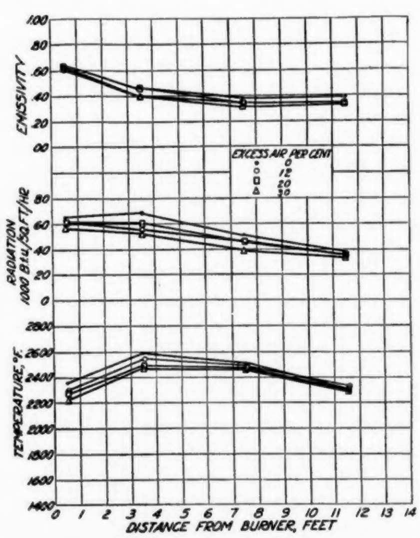

Fig. 15—Temperature, radiation and emissivity of Hocking coal flames with varying excess air, through 200 mesh, 80 per cent; rate of heat input, 2,650,000 B.t.u. per hour

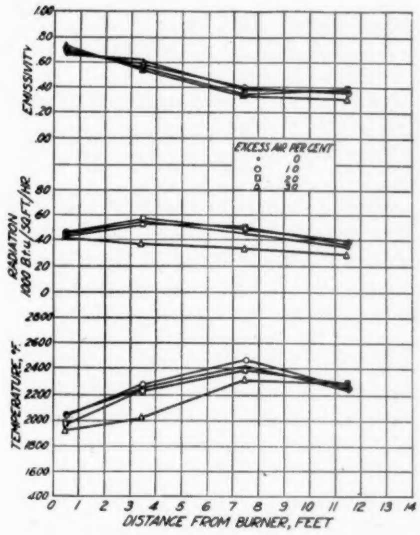

Fig. 16—Temperature, radiation and emissivity of Pocahontas coal flames with varying excess air. Through 200 mesh, 73 per cent; rate of heat input, 2,780,000 B.t.u. per hour

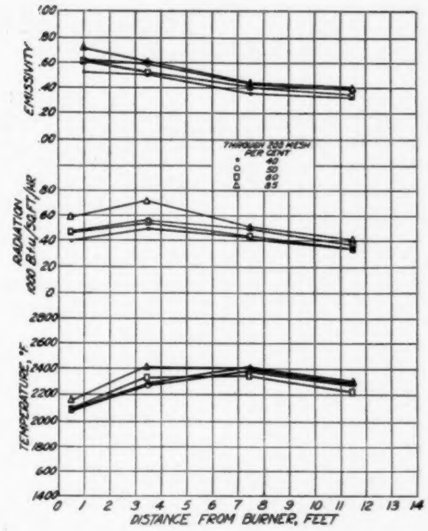

Fig. 17—Temperature, radiation and emissivity of Illinois coal flames, varying percentages through 200 mesh. Excess air 20 per cent; rate of heat input 2,650,000 B.t.u. per hour

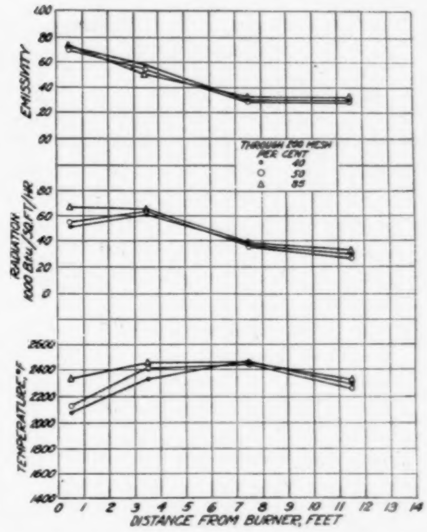

Fig. 18—Temperature, radiation and emissivity of Pittsburgh coal flames, varying percentages through 200 mesh. Excess air 20 per cent, rate of heat input, 2,700,000 B.t.u. per hour

Although the amount of carbon was small at 11.5 ft. from the burner, less than 2 per cent, most of the ash was still suspended in the gas and this, with the CO_2 and H_2O, gave a high emissivity to the flame.

TABLE V—CALCULATED EMISSIVITY OF PULVERIZED ILLINOIS COAL FLAME

Distance from Burner, Ft.	K	Depth of Flame, In. 35	60	120	180	240
0.5	0.036	0.72	0.88	0.99	1.00	1.00
3.5	0.026	0.60	0.79	0.96	0.99	1.00
7.5	0.016	0.44	0.62	0.85	0.94	0.98
11.5	0.015	0.41	0.59	0.83	0.93	0.97

These data indicate that for flames such as these in large furnaces, 15 ft. or more of uniform flame, the emissivities can be taken as 0.9 to 1.0 for the calculation of radiation.

An outstanding conclusion from these measurements is that, because of the variations in temperature, radiation and emissivity along the length of the flame, the problem of the development of a fundamental expression for even the overall transfer of a flame is tremendously difficult and that the problem of estimating the transfer in different parts of a furnace is even more difficult. The latter problem is frequently of more importance than the first because the proper distribution of cooling surface in a furnace would necessitate a knowledge of the temperature and radiation characteristics in various parts of the flame.

The data on the emissivity of the flame for various depths should be helpful in calculations but in addition to the emissivity the temperature of radiation is required. What this should be in a furnace where the temperature rises to a maximum and then falls is not easily determined.

The value of the present results is considered to be principally: (1) That it has been shown that, of the factors investigated, the type of coal had the most effect on the radiation and emissivity and that the size of coal, excess air and rate of heat input were of decreasing importance, and (2) that the measured values with their accompanying data furnish a test for the validity of fundamental or empirical formulas that may be proposed. The test of such formulas by these data is left to their sponsors.

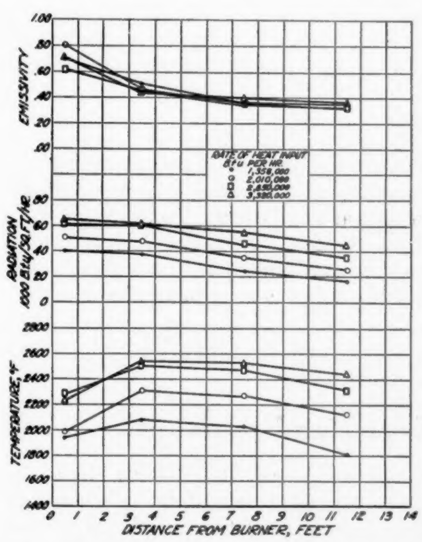

Fig. 19—Temperature, radiation and emissivity of Hocking coal flames, varying rate of heat input. Excess air 20 per cent; through 200 mesh, 80 per cent

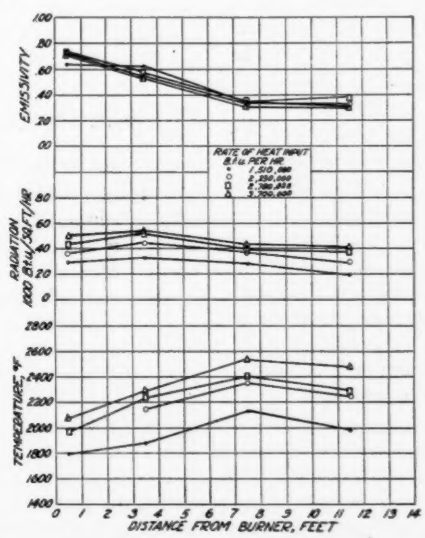

Fig. 20—Temperature, radiation and emissivity of Pocahontas coal flames, varying rate of heat input. Excess air, 20 per cent; through 200 mesh, 70–76 per cent

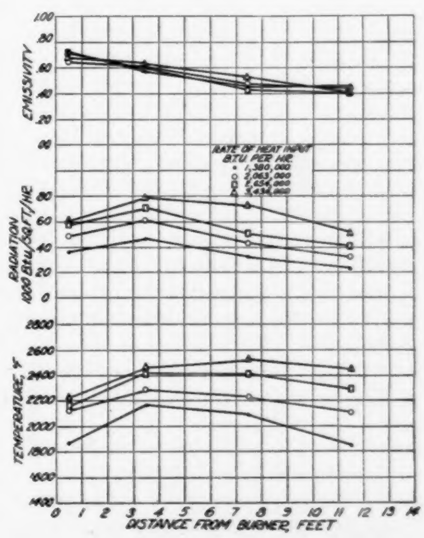

Fig. 21—Temperature, radiation and emissivity of Illinois coal flames, varying rate of heat input. Excess air, 20 per cent; through 200 mesh, 77–85 per cent

Summary

The experimental investigation of the burning characteristics of four coals in pulverized form has shown:

1. That the rate of decrease of unburned carbon with increase of excess air was not the same for different coals. The unburned carbon continued to decrease up to 30 per cent excess air with Pocahontas coal, whereas the curves for Hocking coal were practically flat beyond 20 per cent excess air.
2. That the fineness of grinding becomes increasingly important as the combustion space is restricted and that the optimum limits of fineness, without regard to power consumption in these experiments, differed with the type of coal.
3. That, as shown by the above conclusions, the type of coal markedly influences the rate of combustion. Although the low-volatile Pocahontas coal, 18 per cent on moisture- and ash-free basis, was the slowest burning of the four coals, Illinois coal, volatile content 42 per cent, burned more slowly, and Pittsburgh coal, volatile content 44 per cent, burned only slightly more rapidly than Hocking coal whose volatile content was 40 per cent.
4. That increased furnace temperature increased the apparent rate of combustion of the coals.
5. That, as the difference in the rate of combustion with different coals and furnace temperatures was particularly marked in the early part of the combustion, the ignition temperature of the coal and the temperature in the ignition zone are indicated as important factors in the overall combustion process. The need is evident for a method for the determination of the ignition temperature or relative ease of ignition of pulverized coal when actually suspended in air.

The determination of the radiation from the pulverized-coal flames has shown:

1. That, although the temperature and total radiation of the flame were affected by the fineness of grinding, excess air and rate of firing, the emissivity of the flame at any position was affected to a marked degree only by the type of coal. The maximum differences in emissivities at any position were only 0.10 to 0.15 or about 30 per cent.
2. That the radiation from the suspended carbon and ash particles is an important part of the total radiation for the emissivity of non-luminous gas flame in this furnace was about 0.2, whereas those of the pulverized-coal flames were 0.7 to 0.3, decreasing as the carbon burned from the flame.
3. That a gas flame could be made so luminous, by inducing cracking of the hydrocarbons, that its emissivity could be made greater than that of a pulverized-coal flame.
4. That the absorption coefficient of the flame as calculated from the measurement of the radiation of different thicknesses of flame was not constant because of variable conditions in the flame, but was more constant the greater the thickness and the greater the distance from the burner.
5. That by calculation from the absorption coefficients at the maximum depth of measurement, the emissivities of Illinois coal flames in thicknesses of 15 to 20 ft. would be 0.9 to 1.0.
6. That the variations in temperature, radiation and emissivity along the flame render the problem of developing fundamental expressions for radiant heat transfer in different parts of a furnace, or even for an entire furnace, extremely difficult.
7. That these data furnish a test for the validity of fundamental or empirical formulas for radiant heat transfer.

Hiram Walker & Sons, Inc., have placed an order with The Neckar Company, Inc. for the *Neckar Process* of boiler feed water treatment for their distillery at Peoria, Ill., through their engineers, Smith Hinchman & Grylls. The System will have a capacity to treat 150,000 lb. of raw water per hr.

Stability of Suspension of Coal in Oil

In the September issue of *Fuel in Science and Practice* (England), R. Wigginton relates work carried on by A. B. Manning at H. M. Fuel Research Station on the stability of suspensions of coal in oil.

These investigations showed that petroleum oils, such as paraffin oil and raw fuel oil, whose viscosity does not increase sufficiently with decreasing rate of shear to confer the required stability on coal suspensions therein, can be made to support pulverized coal (85 per cent through 200 mesh I.M.M. screen) by previously dispersing 0.1 to 0.5 per cent of sodium stearate in the oil. The viscosity of the oil under normal conditions of flow is thereby increased, but not unduly. The viscosity at small rates of shear, however, increases rapidly as the rate of shear diminishes, and becomes infinite under the shearing forces involved in supporting small particles of coal.

In other words, toward such particles of coal the dilute gel behaves as an elastic solid and will support them indefinitely. It is not essential that the oil should have a gel structure in order to confer on the coal-in-oil suspension the relative stability requisite for practical purposes (no appreciable settling in six months). This degree of stability may be given by any treatment which affects the viscous properties of the oil in such a manner that its viscosity under low rates of shear has a sufficiently high value, while its viscosity at the rates of shear involved in normal flow remains low enough to give suspensions which can be readily pumped, etc. Mr. Manning concludes that the variation of the viscosity of an oil at low rates of shear is the determining factor from the point of view of the stability of suspensions of pulverized coal therein.

Electric Output

Data on electric output, as compiled by the Edison Electric Institute, show a substantial increase for recent weeks over the corresponding periods of 1932. Although the marked gain for the summer months is not being maintained, September 1933 was 9.7 per cent ahead of September 1932 and the weeks ending November 11 and 18, with outputs of 1,616,875,000 and 167,249,000 kw. hr., respectively, represented gains of 6.3 and 5.6 per cent over those weeks of last year. In all regions except the southern states the output has been greater.

Diesels Removed

A recent announcement in the British press states that two well-known diesel steamships, the "Asturias" and the "Alcantara," belonging to the Royal Mail Steam Packet Company, are going to have the diesels replaced by steam boilers and turbines. It appears that the main objection to the diesels was vibration which was objectionable in a large luxury liner.

A.S.T.M. to Move Headquarters

Headquarters of the American Society for Testing Materials, which for the past fourteen years has been in the Philadelphia Engineers' Club Building, will move at the end of this year to the Atlantic Building, 260 South Broad Street, Philadelphia. The new location will provide more space to accommodate the present staff and the ever-expanding activities of the society. It will also be more accessible to members and visitors.

Power and Heat Cost Analysis for Industrial Plants*

By LOUIS A. OFFER,†

COST analysis in the industrial plant is centered around the articles manufactured and the power, steam, gas and water used is only a part, and often a small part, of the total cost of production. Compared with the total cost it may be that these service items together represent only a few per cent of the total and the conclusion is often reached that they are not of sufficient importance to warrant careful analysis. If their costs, however, are compared with the other main controllable items of expense, such as productive labor, advertising, sales expense, etc., it will usually be found that they are of prime importance and that a complete understanding of them and their proper control may mean the difference between profit and loss.

In most plants the main cost procedure is well established and the final figures indicate the trend of total costs and of some of the principal items that make it up. These are (1) direct materials, (2) direct labor, (3) factory expense, (4) administrative expense and (5) sales expense.

The management has little difficulty in understanding why changes occur in items 1, 2, 4 and 5. Factory expense, however, usually determines the plant's ability to compete, for much depends upon control of the items that make it up.

Probably the greatest number and the most important controllable costs in factory expense can be found under power and heat supply and if these can first be rigorously defined in reasonable and simple terms and then set forth in a manner that can easily be understood, there is little doubt that most of the difficulties in controlling factory expense will disappear.

Lack of progress along these lines is largely because power engineers, plant managers and industrial executives have failed to consider the power and heat that goes into the final products as definite and understandable production supplies. The responsibility for this lack of appreciation rests squarely upon the shoulders of the power engineer for he has in most cases failed to present the relative costs of these items in their proper light to his executives. This inability to constantly interest his superiors is due principally to lack of a code or procedure which would simplify the procuring and recording of quantities and costs and make easily available all required information. What follows is a description of that which the author considers a rigorous and practical method of comparing with production all the important power and heat quantities and cost data up to the point where they are distributed to the production departments.

Monthly Power Reports

There is also a lack of understanding on the part of plant managers as to how often power costs should be analyzed. Some feel that they need to be determined each month only so that they may be added to production costs. Others believe that an analysis should be made occasionally or when added power capacity is needed. Few, however, realize the importance and possibility of savings that can be made by an accurate running record which will serve as a constant guide.

To obtain the greatest value a complete record and analysis should be made at least monthly. If the most important costs revealed are then tabulated with those of previous months and shown on curves, corrections and improvements can be made before valuable time is lost. Weekly or even daily analysis as part of the monthly record can also be used to advantage to stimulate improvements in operation.

Power and Heat Cost Reports

The following record of power and heat costs gives in round numbers the average of figures obtained in about twenty plants. Figures of this kind are available in every accounting department and usually represent quite accurately the totals expended.

POWER AND HEAT COST REPORT—NOVEMBER, 1933

Labor		
Salaries and supervision	$ 1,600.00	
Wages for operating labor	4,100.00	$ 5,700.00
Repairs		
Repair labor	1,100.00	
Repair material	1,400.00	2,500.00
Supplies		
Purchased power	30,000.00	
Coal	18,200.00	
Oil	8,040.00	
Gas	2,000.00	
Water	3,000.00	
Miscellaneous (oil, grease, chemicals, etc.)	560.00	61,800.00
Fixed charges		10,000.00
Total		$80,000.00

Monthly reports of this kind have considerable value but like many flow-meter charts, recording watt-hour meter records, records of coal analysis, etc., they are usually filed away and their potential possibilities wasted. Cost reports to be alive must show the direction and magnitude of the progress being made in lowering manufacturing costs. They must be accurate and be based on records and assumptions that are thoroughly understood and agreed upon.

To show the direction and magnitude of progress, the usual method is to report on the same sheet the results obtained the past month, or the average for the past quarter or that of the same month the year before. Comparisons with these other periods, however, are not always reliable due to the differences in production figures or methods, or to differences in weather conditions which would materially affect the fuel used, or to changes in power and heat-generating equipment or methods.

Power and Heat Records

To be able to judge of a month's power and heat cost, when production and other conditions have changed, it is necessary to break down the total cost figures into the various services so as to discover the location and extent of legitimate changes that should be expected by each change in plant operation. By doing this it is usually simple to establish "par" figures which can be plotted into a sound and practical base line, the deviations from which will indicate progress being made in lowering costs.

Total costs for a whole year shown graphically as in Fig. 1 will give a clear picture of the performance of the power and heat department. The curve at the top shows the amount that the total cost for each month is above or below the par figure established. This curve indicates, by the fluctuation above and below par, that something was wrong with the control of power and heat costs or that the pars were based on unsound analysis. If, however, the variations from par were as shown,

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Fig. 1-a, it could be concluded that after May 1933, control was being practiced, costs were improving and pars were within reason.

Par or Index Power and Heat Cost Values

The following cost and production figures are assumed as an example of what might be found among the back records of a plant.

	Jan. '32	June '32	Nov. '32	Jan. '33
Production—M man-hours	500	750	1,000	1,250
Operating costs	$51,000	55,000	75,000	91,000
Fixed charges	9,000	9,000	9,000	9,000
Total cost	$60,000	64,000	84,000	100,000
Cost per M man-hours	$120.00	85.33	84.00	80.00

If we assume the plant was operated efficiently during these periods, the total and unit figures given should serve as "par figures" and from them we should be able to establish par figures for other conditions. If we consider the two January costs it will be noted that the cost per unit is 50 per cent greater at 500 production than at 1250. This is due to fixed charges and other costs which do not vary with production. These two costs plotted against production will serve as a basis for the Par or Index Chart in Fig. 2 along the lines proposed by Professor Rautenstrauch in his article "Economic Characteristics of the Manufacturing Plants."[1]

TABLE I

	Fixed Charges	Jan., 1932	June, 1932	Nov., 1932	Jan., 1933
Production—M man-hours		500	750	1,000	1,250
Average outside temperature		25°	65°	40°	25°
Production services					
1. Factory power	200	19,000	25,000	31,000	37,000
2. Process heat	4,700	16,000	21,000	27,000	33,000
3. Compressed air	400	3,000	4,800	5,600	6,500
4. Factory water	400	1,000	2,100	2,300	2,400
Factory services					
5. Heating	1,700	8,000	0	5,000	8,000
6. Fire protection and refrigeration	200	500	1,000	500	500
7. Lighting	100	800	300	800	800
Irrelevant services					
8. Power and heat sold	1,300	2,700	800	2,800	2,800
Total costs	9,000	51,000	55,000	75,000	91,000

If, however, the June cost is plotted on the same chart, it will fall considerably below the total cost line established for January. This is due to the elimination of all or most of the costs that are caused by weather conditions such as heating and extra lighting. These weather condition costs are usually quite independent of production rates and may be considered as costs that are fixed according to the outside temperature. If this latter assumption is correct a line drawn through the June cost parallel to the January line will represent the summer total cost for all production rates. The total cost at zero production determines the summer total fixed cost line as in Fig. 2.

Parallel lines may now be drawn between the summer and mid-winter ones so as to represent the costs for different months or different outside temperatures. The "par" or index can now be found for any combination of production and

[1] Mechanical Engineering, Nov. 1932, page 759.

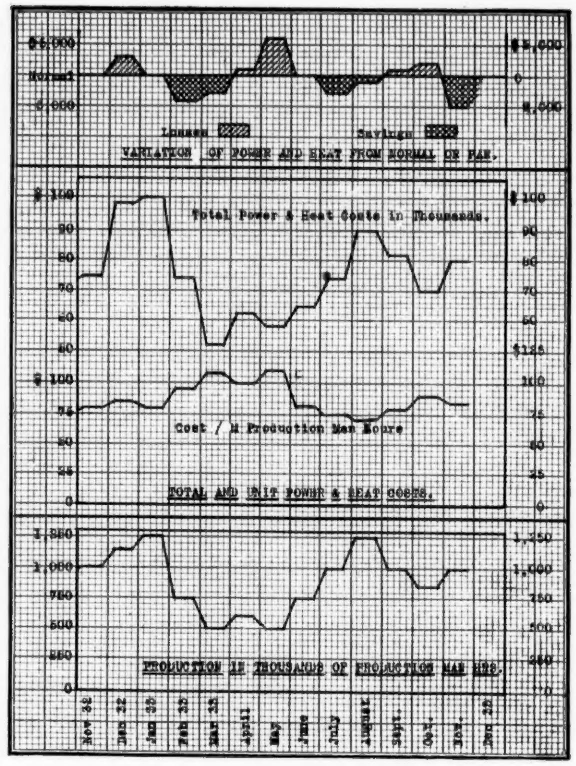

Fig. 1—Total costs for year

November 1933

TABLE II—POWER AND HEAT COST ANALYSIS SHEET

		T	S	P	W	A	B	C1	C2
			Generated Supplies			Production Services			
						Factory	Process	Compressed Air	
Group 1	Major Supply Balances	Totals	Steam	Power	Water	Power	Heat	Steam	Electric
1	Live steam, M lb.	100,000	5,000	20,000	5,000	0	60,000	4,000	0
2	Exhaust steam (credit −), M lb.	0	5,000	−15,000	−4,500	0	0	−3,500	0
3	Total steam, M lb.	100,000	10,000	5,000	500	0	60,000	500	0
5	Generated power, kw. hr.	420,000	0	20,000	20,000	380,000	0	0	0
6	Purchased power, kw. hr.	2,000,000	0	0	0	1,620,000	100,000	0	120,000
9	Service water, M gal.	100,000	0	0	0	0	0	0	0
10	Purchased water, M gal.	30,000	4,000	200	0	0	0	4,000	4,000
Group 2	Production and Service Units	M man-hr.	M lb.	kw-hr.	M gal.	kw-hr.	MM B.t.u.	M cf. F. A.	M cf. F. A.
11	Total units produced	1,000	90,000	400,000	100,000	2,000,000	90,000	40,000	40,000
12	Service units per man hour	0.001	90 lb.	0.4 kw-hr.	100 gal.	2 kw-hr.	90 MM B.t.u.	40 cu. ft.	40 cu. ft.
Group 3	Labor and Supply Costs	$	$	$	$	$	$	$	$
15	Salaries and supervision	1,600.00	500.00	200.00	50.00	200.00	190.00	50.00	120.00
16	Wages for operating labor	4,100.00	2,000.00	450.00	50.00	450.00	210.00	100.00	380.00
17	Repair labor	1,100.00	350.00	150.00	80.00	200.00	50.00	50.00	30.00
18	Repair material	1,400	700.00	230.00	50.00	170.00	0.00	50.00	50.00
20	Purchased power (6 x 1.5¢)	30,000.00	0.00	0.00	0.00	24,300.00	1,500.00	0.00	1,800.00
21	Coal, oil and gas	28,240.00	18,200.00	0.00	0.00	0.00	10,040.00	0.00	0.00
22	Purchased water (10 x 10¢)	3,000.00	400.00	20.00	0.00	0.00	0.00	400.00	400.00
23	Oil, grease, chemicals, etc.	560.00	350.00	50.00	25.00	0.00	0.00	25.00	20.00
24	Gen. steam dist. (3 x 29S)	x	22,500.00	1,250.00	125.00	0.00	15,000.00	125.00	0.00
25	Gen. power dist. (5 x 29P)	x	x	2,400.00	120.00	2,280.00	0.00	0.00	0.00
27	Service water dist. (9 x 29W)	x	x	x	500.00	0.00	0.00	0.00	0.00
28	Total operating cost	70,000.00	x	x	x	27,600.00	27,000.00	800.00	2,800.00
29	Operating cost per net service unit	70.00	0.25	0.006	0.005	0.0138	0.300	0.020	0.070
Group 4	Fixed Charges								
31	Taxes, ins., int., dep. and obs.	10,000.00	7,200.00	400.00	300.00	150.00	350.00	350.00	200.00
32	Dist. of steam charges in 32S	x	7,200.00	400.00	50.00	0.00	4,350.00	50.00	0.00
33	Dist. of gen. power F. C. in 33P	x	x	800.00	50.00	750.00	0.00	0.00	0.00
35	Dist. of ser. water F. C. in 35W	x	x	x	400.00	0.00	0.00	0.00	0.00
36	Total fixed charges	10,000.00	x	x	x	900.00	4,700.00	400.00	200.00
45	Total costs (28 + 36)	80,000.00	(30,700.00)	(3,200.00)	(900.00)	28,500.00	31,700.00	1,200.00	3,000.00
55	Total cost per service unit	80.00	0.330	0.008	0.009	0.0143	0.3522	0.030	0.075

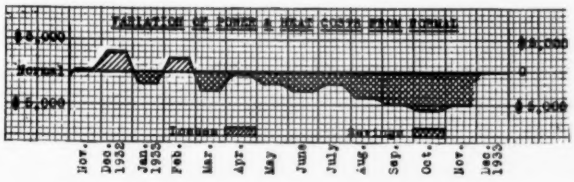

Fig. 1a—Variations of costs from normal

weather conditions. Results obtained during a few months after establishing the "Par Chart" will soon indicate whether or not the assumptions made were sound and a new chart can be made if necessary.

Par values based on records of total costs only, will serve to indicate the variation of total costs from normal but fail to indicate why and where there has been a failure or an improvement that has caused a change from normal. If, however, total costs are broken down into the several services that make up the power and heat services of the plant, as shown in the following table, separate charts or sets of index figures can be established which will indicate just where improvements or failures have occurred.

Fig. 3 shows the total operating costs for each of the production services given in Table I, plotted against production. If these vary directly with production a straight line can be drawn through them all as shown for factory power. If, however, all those for the warmer months fall on the same side of the straight line drawn through the costs during the colder months it is an indication that outside temperature is also affecting the cost. This condition is indicated by the points plotted for June for process heat, compressed air and service water. These variations can and should become marked especially for compressed air, for power can be saved by colder intake air and less cooling water will be required.

Fig. 4 shows the various factory service costs of Table I. The curve for heating is established by computing values for the months whose costs are not known, by using the ratios of the degrees below 65 fahr. (assumed inside temperature) that occurs each month.

The fire protection cost is assumed constant for all months and refrigeration varies as the temperature increases above 65 fahr. Lighting costs are varied according to the daylight available.

With curves available for each service as in Figs. 3 and 4,

D	H	JK	L	M
	Factory Services			
Factory Water	Heating	Fire Protec. and Refrig.	Lighting	P & H Sold St. Power and Water
0	4,000	0	0	2,000
0	14,000	0	0	4,000
0	18,000	0	0	6,000
0	0	0	0	0
0	0	20,000	20,000	120,000
80,000	0	2,000	0	18,000
17,300	0	500	0	0
M gal.	M lb.	Tons	kw-hr.	
87,300	18,000	500	20,000	1,000
87 gal.	18 lb.	1 lb.	0.02 kw-hr.	
$	$	$	$	$
50.00	100.00	30.00	50.00	60.00
80.00	170.00	60.00	50.00	100.00
0.00	30.00	0.00	50.00	100.00
0.00	0.00	0.00	50.00	50.00
0.00	0.00	300.00	300.00	1,800.00
0.00	0.00	0.00	0.00	0.00
1,730.00	0.00	50.00	0.00	0.00
40.00	0.00	50.00	0.00	0.00
0.00	4,500.00	0.00	0.00	1,500.00
0.00	0.00	0.00	0.00	0.00
400.00	0.00	10.00	0.00	90.00
2,300.00	4,800.00	500.00	500.00	3,700.00
0.026	0.267	1.000	0.025	3.700
50.00	150.00	200.00	100.00	550.00
0.00	1,650.00	0.00	0.00	700.00
0.00	0.00	0.00	0.00	0.00
350.00	0.00	0.00	0.00	50.00
400.00	1,800.00	200.00	100.00	1,300.00
2,700.00	6,600.00	700.00	600.00	5,000.00
0.030	0.366	1.400	0.030	5.000

accurate "operating par" cost figures can be quickly established by adding the separate operating pars of the services. If we add to this par the fixed charges and the cost of the services used in other departments or plants which had nothing to do with the plant in question, we will obtain the total par. These latter costs are designated as Power and Heat Sold in Table I.

Power and Heat Bookkeeping Responsibility

If the power engineer hopes to educate the executive in the economics of steam and power he must make up his mind to take over the major portion of the bookkeeping involved, for the most important items of cost are seldom, if ever, fully appreciated or understood by those in any other department. In the smaller plants the chief engineer should keep complete records of all quantities used and distributed, and compute the cost of producing the various services. By doing so he will have a better understanding of the relative costs of different methods used and be able to direct operations more efficiently. In the larger plants a technically trained clerk will perform these duties for him.

In the larger plants the services of a high-grade technically trained engineer as a "power and heat despatcher" should be seriously considered. The services of such an engineer and the savings he was able to accomplish with the aid of a well-worked out cost system is described in a paper by H. M. Burke in the 1923 *Transactions* of the A.S.M.E. The duties of the power despatcher are given therein as follows:

"*a.* Inspection of all distributing lines and conductors (except sprinkler system), and all valves, traps, appurtenances, and instruments thereon, for the purpose of preventing any losses of heat, light and power in transmission.

"*b.* Investigations and studies of the manners in which steam, water and power are used for various purposes, with the aim of devising and recommending a more economical practice.

"*c.* Recording of steam, power and water consumption by various departments and preparation of reports and cost charges."

Power and heat costs accumulated or presented in an indifferent manner, by chief engineers, technically trained power

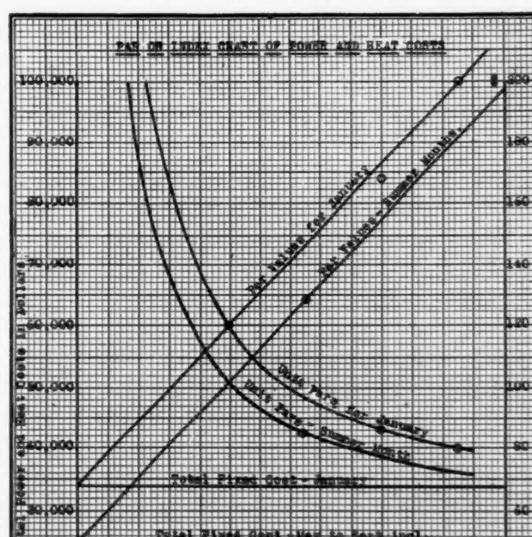

Fig. 2—Par or index chart

cost clerks or high-grade power despatchers are usually of less value than those produced by cost or accounting departments. If, however, these men who deal directly with the power and heat production and distribution are supplied with the proper tools in the form of explicit power cost procedures, standards for crediting and debiting quantities and reliable, though simple, measuring apparatus, they can produce figures which will be of real service in controlling costs. The first and most important tool required is a monthly breakdown form that will guide the costs into their proper places. Table II illustrates, and what follows describes briefly, a form that has been found to fit all types of plants.

Arrangement of Breakdown Sheet

A special breakdown based on engineering considerations rather than an accepted accounting procedure alone is required so that the following requirements may be fufilled.

1. Total quantity of the various purchased and generated supplies used by each service.
2. Total quantities of units produced in each generated supply and service must be shown so that a unit cost can be determined and so that comparison can be made with quantities used during other months and at other plants.
3. Detailed and total cost of each generated supply and service must be shown so as to be able to determine unit costs to be used in charging for the number of units used by other departments. The detail, total and unit costs must be so arranged that they may be easily compared with those found in other months or in other places.

The first step in the production of a breakdown that will fulfil these requirements is the selection of a sheet that will provide one column for totals, one for each generated supply and one for each power and heat.

Power and Heat Services

As stated previously, and as shown in Table 1, the services are divided into three classes, namely, production services, factory services and irrelevant services. The first class is divided into the following: A—Factory power; B—Process heat; C—Compressed air; D—Factory water; E—Refrigeration for process and G—Hydraulic service.

Factory power includes all of the power, both purchased and generated, used by the plant. This may be in the form of electrical energy or direct power furnished by steam engines, gas engines or hydraulic turbines, water wheels, etc. Wherever possible it should include only the energy used to drive machinery but in special cases it is sometimes allowed to include the electrical energy used for process heating and for lighting. The inclusion of these latter items should be discouraged. The electrical energy for process heating belongs to the second class where its cost can be compared with other means of heating materials whether it is for heat treating, welding or warming water. The electrical energy for lighting belongs in the third class since the amount required depends on demands that have little or nothing to do with power requirements.

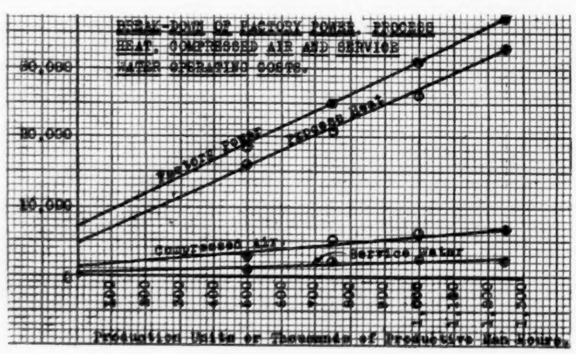

Fig. 3—Service costs plotted against production

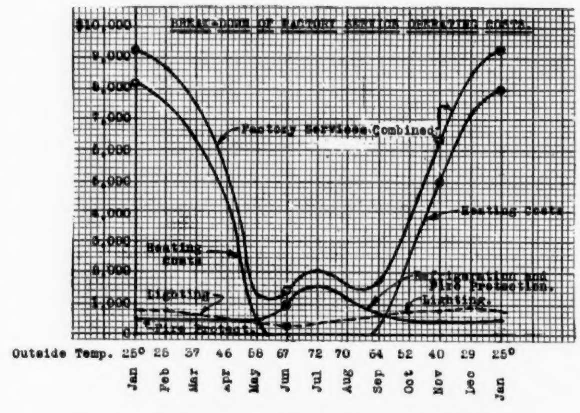

Fig. 4—Factory service costs from Table I.

Process heat includes the electrical energy, the oil, the gas and the steam that is used to heat process work, water, etc., but does not include steam, fuel or electrical energy to heat or light the plant. These latter items vary with the seasons or other special conditions.

Compressed air includes all of the air compressed in the plant. If both steam and electric compressors are used the service should always be split, that is, C-1 compressed air steam and C-2 compressed air electric, so as to show which is the more economical.

Factory water includes all water used in the plant.

Refrigeration should also be split up if different types of machines are used. Where refrigeration is used for drinking water or for building cooling it should be included in the second class, namely, factory service.

Hydraulic service may need to be split up according to the pressure and for types of drive used.

The second class contains the items that are more or less independent of production. The following subdivisions allow for a rather detailed split-up which will discourage carelessness and the ignoring of easily wasted commodities: H—Factory heating; I—Factory ventilation; J—Fire protection; K—Drinking water refrigeration and L—Factory lighting.

The third class covers all of the various power and heat items not used in the plant under consideration. It includes the steam, water and power sold or delivered to other plants of the company without charge. Its designation is, M—Power and heat sold.

Name Space Titles

Ordinarily the name space titles of a cost breakdown sheet include only the cost items. This requires reference to other sheets in checking back or in trying to account for the size of a charge for a supply. This difficulty has been eliminated by placing at the top of the sheet complete balances of all of the major supply items as shown in Group 1, Table II.

Group 2, Production and Service Units, serves the purpose of fixing the number of net units available for distribution. It also shows the relation between the number of service units in each generated supply and service, and the production units of the plant.

Group 3, Labor and Supply Costs, covers all the labor and supply costs and a distribution of the generated supplies. The method of distribution is indicated by the formulas and the heavy black line.

Group 4, Fixed Charges, includes the fixed charges on power and heat equipment. Line 31 covers the total charges while lines 32 to 35 cover the distribution of the generated supply charges which are carried down from above.

It should be noted that fixed charges are distributed on the basis of demand while the operating costs are distributed according to quantities used as shown in the balances.